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The Theory and Practice of Heating and Ventilation

Arthur H. Barker.

THE THEORY AND PRACTICE
:: :: :: :: OF :: :: :: ::
HEATING AND VENTILATION.

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BARKER

ON

HEATING.

THE THEORY AND PRACTICE
OF
HEATING AND VENTILATION

BY

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PREFACE.

THE object of the present work is to place before English speaking readers a detailed explanation of the application of scientific principles to the chief problems with which the Heating Engineer has to deal, and further, to indicate the limits within which these principles are applicable in practice. Its aim is not to describe practical details of construction nor of the mechanical work involved in the erection of an apparatus. These, in the opinion of the Author, can only be learnt properly by handling the things themselves.

Considerable experience of this, as well as of several other important branches of engineering, has convinced the Author that apparent discrepancies between theory and practice are here both more important and more difficult fully to account for than is the case in most other branches. A full comprehension of the limitations within which theory is applicable is therefore almost as necessary to a thorough understanding of the subject as knowledge of the principles themselves.

There is a marked tendency, even in the best modern practice, to accept blindly the dicta of experimental and theoretical science as directly applicable to the solution of problems in heating and ventilation without any inquiry as to how far the conditions of practical work warrant this implicit faith. The fact that the validity of some of these applications has not more often been called in question is probably due to the extraordinary difficulty of making exact practical tests to verify or refute it.

A critical examination of the applicability of scientific principles to each part of this subject, made in the light of practical experience, has therefore been ventured on in this work.

The theoretical part of the book is intended for readers who possess a sound though elementary knowledge of mathematical science. It is hoped that the practical parts will enable those who are not very familiar with the theory readily to apply the results in practice.

The problems of heating and ventilation are amongst the most subtle and difficult of accurate solution of any with which an engineer has commonly to deal. It is not possible to explain intelligently and accurately the application of fundamental principles to their solution, in a manner which is wholly free from the assumption of a certain amount of scientific knowledge of an elementary character. Any attempt to do so must inevitably lead to the making of inaccurate statements and conveyance of false impressions.

In the earlier part of the book, therefore, is given a *resumé* of the leading scientific principles with which an acquaintance is assumed. This part of the work is merely a brief account of those sections of physical science, good knowledge of which is, in the opinion of the Author, the bare minimum necessary for a thorough understanding of this subject. Readers for whom this *resumé* is not sufficient are recommended to study elementary text books on the respective subjects touched upon.

Complete theoretical solutions of most of the problems dealt with by the heating engineer are so complicated that only experimental investigations are of practical utility. Even the results of these, howsoever accurately made, cannot safely be used without careful scrutiny of the conditions under which the experiments were made, and a comparison with the practical conditions to which the results are to be applied.

The results of the work of different experimenters have been in many cases so divergent that it is often difficult not only to reconcile them, but even to deduce from a comparison between them any approximate rules for the guidance of the engineer. Recorded results of experiments are so numerous that it is impossible to indicate even the outlines of the best work that has been done without extending this volume to an inordinate length.

Further, it is the experience of the Author, that to provide a mass of divergent experimental results only serves to confuse the practical man, for whom this book is primarily intended. It has therefore been thought advisable in each case to give what the Author believes to be the best results and methods only, without explaining, or in many cases even alluding to the existence of other methods and results, which he thinks not so generally useful to the heating engineer. The particular values selected are in all cases those which he knows from practical experience to be useful, satisfactory, and correct, within certain limits, which he has endeavoured clearly to indicate.

There are certain parts of mechanical science especially applicable to the work of the heating engineer, which receive in most text books very scant and inadequate treatment, such that they cannot be thoroughly grasped without prolonged study even by fairly advanced students. In particular, such subjects as the flow of fluids, the conduction of heat, the theory of fans, and even the principles of the ordinary low pressure circulation of water and of air are matters of which it is very difficult to find any convincing elementary explanation in any language. Special attention has therefore been devoted to these points.

The geometrical proofs of the principles of the theory of hot water circulation and the method of pressure diagrams, given in Chapters XXII. and XXIII., are here published for the first time and are believed to be quite new.

The general neglect in English practice of the rational method of determining pipe sizes, and the entire absence from English text books of any explanation of the fine work done, in the first place by Rietschel, seem to justify the earnest attempt which has been made in the present volume to fill up this serious gap in the literature of this subject. But the minuteness of detail which that able writer uses seems to be somewhat unsuitable for use in this country, and a simplification of his method has been devised by the present Author, who has also made a very extensive series of experiments which are not yet completed, with the object of testing the applicability to English practice of the Weisbach co-efficients used by Rietschel.

The graphical method of treatment, which appeals strongly to the present writer, has been adopted where it has been thought suitable.

It has not been possible to give in all cases the authorities from which the tables and results have been originally extracted. Many of them are taken from the Author's private note book, which has been compiled at various times during the past 12 years without any idea of publishing its contents, and he has not in all cases made a note of the source of the information. Wherever the source is known to himself, however, he has quoted it. Many of the values adopted are those of Rietschel, which are in general use on the Continent.

In most cases the tables have been calculated by the Author, or have been obtained from his own personal experience and from the large number of experiments he has himself conducted on a practical scale. With one or two exceptions, the whole of the numerous diagrams have been calculated and plotted by himself.

The question of the units to be adopted has received careful consideration. There is a strong temptation in the preparation of a work dealing mainly with the scientific aspects of a technical subject to abandon entirely the tangle of absurd units with which the English speaking races are afflicted. It was felt, however, that though the adoption of this course would have considerably lightened the great labour involved in the preparation of this work, it would probably have formed an obstacle to the general adoption of the principles and methods herein recommended.

For good or ill the pound, foot, and inch are, and are likely to remain, with us. These and units derived from them have been adopted in all the tables, but tons, cwts., quarters, chaldrons, ounces, grains, yards, gallons, ought to be swept away along with rods, poles and perches, and similar antiquities. Such units are not used in this book, and the Author ventures to recommend Heating Engineers not to tolerate them at all, except so far as contact with other trades makes their occasional use unavoidable. It is to be hoped that their use in commercial life will before long be abolished by legislation.

The unit of pressure adopted for low pressure steam heating is here the inch of water column at 200° F., or 5 pounds per square foot, which the Author has found, and believes that others will also find, a much more convenient unit in this connection than the pound per square inch. For certain appropriate purposes the inch of mercury column, and the foot of air column and the pound per square inch are also employed.

It is not possible for anyone to compile a treatise on the science of heating and ventilation without deriving many ideas from the admirable work done by Rietschel, Weisbach, Peclet, Recknagel, and other foreign and American engineers. The work of these scientists is so well known, and, except in this country, in this particular branch of engineering, so universally adopted that it has come to be part of the common knowledge of the engineer, and any specific acknowledgment is in many cases possibly unnecessary, and for the reasons stated above, in the present case impossible.

It is hoped that in this volume will be found most of the scientific and practical information which the heating engineer requires in his ordinary practice, and that the reasoning on which the information is based will be found explained in a manner which embodies both sound science and sound common sense—two points of view which it is often difficult to reconcile—and especially so in this subject.

The Author wishes to express his thanks to Dr. Leonard Hill, of the London Hospital, for kindly furnishing him with some valuable information on the results of recent researches by Dr. Hill himself and others, on the physiology of ventilation, which is contained in Chapter VI.

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Queen Anne's Chambers,
Westminster, S.W.,
1912.

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CHAPTER I.

THE PROPERTIES OF MATTER, MOTION, AND ENERGY.

THE practical carrying out of the ventilation and heating of a building consists essentially of delivering to different parts of the building a constant supply of fresh air and of heat.

The transference of air from the outside to the inside of the building is a direct process, but the communication of heat is generally an indirect one. It consists of (1) the generation of the requisite quantity of heat at a particular point in the building or out of it, (2) the communication of the heat to fluid, (3) the transference of the warm fluid from the place where the heat is communicated to it to the spot where the heat is required, (4) the communication of the heat of the fluid to the apartment, and usually (5) the subsequent return of the cooled fluid to the furnace or source of heat, in order to be re-heated, so as to be made available to carry a fresh supply.

The whole of the process, both of heating and ventilation, is one involving the movement of fluids and the application of the properties of heat. It is therefore necessary, as a preliminary step, to give a brief summary of the laws of mechanics on which the motion of fluid or any other body fundamentally depends, and of the laws of heat.

The bases of all the laws of mechanics are Newton's laws of motion, which may be briefly stated as follows :—

All matter persists in the state of rest or motion in which it happens to be, unless acted on by some external force, and the rate of change of motion is proportional to the force impressed upon the matter.

The quantity of matter in any body, solid or fluid, is known as its "mass." The practical unit of mass in English speaking countries is one pound.

The "mass" of a body is an essentially different thing from its "weight." The mass is the *quantity of matter* in a body. Its weight is the *force* with which that mass is attracted by the earth.

A clear conception of the difference between these two ideas is necessary to a proper understanding of all mechanical science. The masses of any two bodies are always proportional to the weight of those bodies at a given spot.

The mass of a body is invariable throughout the universe. Its weight is variable and depends on the proximity of other masses. Thus, the weight of the same body when measured by a delicate spring balance varies slightly at different places on the surface of the earth, according to the proximity to the centre of the earth.

It is an essential property of all matter that there is a natural force of mutual attraction between any two masses which is proportional to their product, and inversely proportional to the square of the distance between them. This force of mutual attraction is always the same between any two given masses at a given distance apart, throughout the universe. One example of the existence of this force is the *weight* of a body, which is the mutual attraction between the mass of a body and the mass of the earth. Thus, the weight of a given body is not constant, but depends on the magnitude of the other mass and its proximity to it.

The other two fundamental units are the unit of time—the second, and the unit of length—the foot.

Unit of Velocity.—When a body moves relatively to other bodies at a certain speed, the speed or "velocity" is measured by the number of units of distance traversed in one unit of time. The unit of velocity is therefore one foot in every second, known in mechanics as one "foot per second."

Unit of Momentum.—The momentum of any moving body is the *amount of movement* in it. The unit of momentum is the amount of movement in one unit mass (one pound) moving with unit velocity (one foot per second). The amount of movement or momentum of any moving body is the product of its mass into its velocity. Thus, when a mass of half a pound is moving at 15 feet per second, the amount of movement in it is $\frac{1}{2} \times 15 = 7\frac{1}{2}$ units of momentum.

The "*unit of acceleration*" is a *rate of increase* of velocity, such that the velocity of the moving body in question increases by one unit of velocity every second. Thus, if at the end of one second it is moving at 3 feet per second, and at the end of the next second at 4 feet per second, and so on, it has one *unit of acceleration,* known as one "foot-per-second per second."

Unit of Force.—One essential property of any mass of matter is that when a constant force acts on it, it produces an alteration in the amount of movement in it. This increase is manifested as a constantly and regularly increasing velocity. The accurate measure of any force is the *amount of movement* which that force generates when it acts during a unit of time. Every absolute unit of force will produce one unit of momentum for every second during which it acts on any body perfectly free to move in the direction in which the force acts. Thus, one unit of force will produce in one unit of mass, one unit of velocity in one unit of time, or in half a unit of mass two units of velocity in one unit of time, and so on.

The velocity produced is proportional to the time, thus, one unit of force will in 3 seconds produce 3 units of velocity in one unit of mass, *i.e.*, a velocity of 3 feet per second in a mass of one pound. Expressed in another way, the unit of force produces unit "acceleration" in unit mass.

This unit of force, known as the "poundal," is not used in engineering practice, except for purposes of explanation of fundamental scientific facts. Accurate measurements show that the force which the earth exerts on any mass (take for example a mass of 2½ pounds) is sufficient to produce in that mass, whatever its magnitude may be, a rate of increase of velocity, or an "acceleration" of 32.2 feet per second every second. This quantity, which is known in mechanical science as "g," is the "acceleration due to gravity." It is thus clear that the force which the earth exerts on every pound of matter is 32.2 *absolute units of force,* for it produces an acceleration of 32.2 feet per second per second in it. For engineers, the important fact to observe is, that the *weight* of a body acting on its *mass* will produce a rate of increase of velocity of 32.2 units every second *whatever the mass may be.* It is found far more convenient for engineers to use the slightly variable weight of one pound as the *practical* unit of force, instead of the theoretical and absolutely constant force of one poundal. The latter is at a rough approximation about equal to the weight of half an ounce.

1/32.2 of 1 lb. weight.

The drawback to the use of the pound weight as a unit of force is that it makes difficult of comprehension many of the fundamental scientific truths with which every engineer should be very familiar.

The fundamental equation relation to this matter is as follows :—

The force in poundals = the mass in pounds on which it acts × by the velocity in feet per second, which the action of the force generates in every second during which it acts. In brief this equation is written

Force = mass × acceleration, or

$$f = m \times a \qquad (1)$$

Laws of Moving Bodies.

If a body moves with a velocity u feet per second for t seconds, the distance it travels during that time is obviously

$$s = u \times t \qquad (2)$$

that is, if the number of feet per second is multiplied by the number of seconds, the result is the distance travelled in the time.

If the velocity with which the body moves is not constant, as in the case of a falling body (whose velocity increases regularly with the time), the distance travelled is the *time average of the velocity* multiplied by the time.

Thus, if a body is thrown downwards with a velocity u feet per second, its speed is being continually increased by the earth pulling on it by 32.2 units of velocity every second. It has at the beginning of the time during which it is falling a downward velocity of u feet per second. At the end of the time t seconds it has a downward velocity of

$$v = u + (g \times t) \text{ ft. per second.} \qquad (3)$$

Since the amount of increase of velocity is proportional to the time, its *average* velocity through the interval is

$$\frac{u + (u + g\,t)}{2} \text{ ft. per second} = u + \frac{g\,t}{2} \qquad (3a)$$

Therefore the space through which it travels is

$$S = t\,(u + \frac{g\,t}{2}) = u\,t + \tfrac{1}{2}\,g\,t^2 \qquad (4)$$

Squaring equation (3) and rearranging we have

$$v^2 = u^2 + 2\,g\,(u\,t + \frac{g\,t^2}{2})$$

or substituting from equation (4)

$$v^2 = u^2 + 2\,g\,s \qquad (5)$$

For the heating engineer the most important of these formulæ is No. 5, which shows that if a body falls from rest (*i.e.*, $u = o$) through a height h feet, the velocity it acquires is

$$\sqrt{2\,g\,h} = 8.025\,\sqrt{h}$$

or about $8\,\sqrt{h}$

This formula is constantly recurring throughout the whole of this treatise, and the student is advised to make himself thoroughly familiar with it, otherwise it is impossible for him to understand properly the science of heating and ventilation, or indeed any other part of mechanical science.

Work in the scientific sense is performed when any force acts on a body and pulls it in the same direction as the force acts through any distance. The measure of the work is the product of the force and the distance.

Thus, if a horse pulls a carriage along a road over a distance of 50 feet, exerting all the time a force of 150 pounds, the work it does during the process is

$$150 \times 50 \text{ foot pounds.}$$

The absolute unit of work is the absolute unit of force (one poundal) acting through the absolute unit of distance (one foot). This unit is called the "foot-poundal," but as the poundal is not used by engineers as the unit of force, but the pound weight instead, which is about 32.2 times larger, the "foot-pound" is used as the practical unit of work.

"Energy" is the *power* of doing work. The amount of energy which any given condition confers upon a body is the maximum amount of work which that condition will enable it to perform under suitable circumstances. The unit of energy and the unit of work are therefore the same. Energy is work *in posse*. Mechanical work is the same thing *in esse*.

A body may possess energy or the power of doing work in virtue of a number of different properties or conditions which it may have.

(1) Energy due to its position, called "potential energy."
(2) Energy due to its motion, called "kinetic energy."
(3) Energy due to the heat it contains, or "heat energy."
(4) Energy due to its chemical properties, called "chemical energy."
(5) Energy due to its electrical condition, or "electrical energy."
And many other kinds of less importance to the engineer.

All energy of whatever kind is essentially of the same nature. The meaning of the statement that any form of energy is identical with any other form is that energy of one kind can be converted into energy of any other kind without any by-products. Energy is of its very nature indestructible. It can only be converted into an equivalent amount of some other kind of energy, and can never be annihilated. There is a tendency for every form of energy to degrade into the form of heat.

Every well defined quantity of energy of every known kind can therefore be measured in terms of the unit of work, *i.e.*, the foot-pound.

(1) *Potential Energy.*—This expression is used especially with reference to the energy which a body has due to its position relative to the surface of the earth. Thus, if a body, weighing say 12 pounds, is raised through a distance of 20 feet further from the centre of the earth, the additional amount of potential energy thereby conferred upon it is

$$12 \times 20 = 240 \text{ foot-pounds.}$$

This quantity of work can be recovered from the body by allowing it to descend through the same distance.

(2) *The Kinetic Energy* of a body is its power of doing work owing to the velocity which it has relatively to the earth, or any other relatively fixed mass. The magnitude of the energy which a body of m pounds possesses, due to moving at a velocity of v feet per second is

$$\tfrac{1}{2}\, m\, v^2 \text{ foot-poundals}$$

or expressed in foot-pounds

$$m \frac{v^2}{2\,g} \qquad (6)$$

Proof of this is as follows :—

If a force of f poundals acts on a mass of m lbs., it produces acceleration of

f/m ft. per second per second. (See Eq. 1.)

After this force has acted through a distance of s ft. the velocity of the body will be

$$v = \sqrt{2\frac{f\,s}{m}} \text{ ft. per second.} \quad \text{(See Eq. 5.)}$$

From the principle of the conservation of energy the body moving at that velocity obviously must contain all the energy which has been communicated to it by the force of f units acting through a distance of s feet, *i.e.*, a body m pounds moving with a velocity of

$$v = \sqrt{2\frac{f\,s}{m}} \text{ ft. per second}$$

contains

Kinetic energy = $s \times f$ foot-poundals,

but

$$s \times f = \left(\sqrt{2\frac{f\,s}{m}}\right)^2 \times m \times \tfrac{1}{2} = \tfrac{1}{2}\, m\, v^2 \text{ in foot-poundals.} \quad (7)$$

To reduce this to foot pounds, it is clearly necessary to divide by g, since there are g poundals in one pound weight.

This is a formula of the greatest possible importance, as it applies, in addition to solid bodies, to air, water, steam, or any other fluid with which the heating engineer may have to do.

For instance, find the amount of kinetic energy in 1,500 cubic feet of air at 80° F., when moving at a velocity of 13 ft. per second. The calculation is given below :—

Weight of 1,500 cubic feet of air at 80° (see Table V.).

$$= 73.6 \times 1.5 = 110 \text{ lbs.}$$

Kinetic energy of 110 lbs. moving at 13 ft. per second

$$= \frac{110 \times (13)^2}{2\,g} = 290 \text{ foot-pounds.}$$

(3) *Heat Energy.*—Heat is itself a form of energy which is dealt with in the next chapter. Here it will suffice to say that one British thermal unit is the exact equivalent of 778 foot-pounds of mechanical work.

(4) *Chemical Energy.*—For the purposes of the heating engineer chemical energy can best be measured in terms of the unit of heat. For instance, the chemical energy of one pound of a certain quality of coal may be expressed as the equivalent of 14,000 British thermal units, and this again is the equivalent of 14,000 × 778 foot-pounds.

(5) *Electrical Energy.*—It is not, of course, possible within the scope of the present work to give an account of the outlines of electrical science, for which the reader is referred to separate treatises, but it is necessary for a heating engineer to be familiar with the use of two or three of the principal formulæ whereby he can convert electrical energy into heat energy.

There are three practical electrical units in which all calculations are worked out, and on which all other electrical units depend.

1. The unit of current, known as the ampere.
2. The unit of resistance, known as the ohm.
3. The unit of electrical pressure or electro-motive force, known as the volt.

These units are based on absolute standards, the nature of which it is not necessary for present purposes to explain.

The ampere is one-tenth of the absolute unit. The volt is one hundred million times as great as the absolute unit, and the ohm is one thousand million times as great as the absolute unit of resistance.

For the engineer it is necessary to be familiar with the fact that if an electrical difference of pressure of E volts is maintained by a dynamo or otherwise, *on the terminals* of a resistance of R ohms, the current in amperes which will flow through the resistance equals the number of volts E divided by the resistance, or

$$C = E/R \tag{8}$$

When a current of C amperes is maintained by a pressure of E volts, there is a continuous dissipation of energy in the form of heat, proportional to the product of the volts and amperes that is, *rate of dissipation of energy* $\propto C\,E$.

But from the equation

$$C = E/R$$

we have

$$C\,E = C^2\,R$$

so that the rate of production of heat units may be calculated either from the formula $C\,E$ or $C^2\,R$.

In order to convert these symbols into actual figures, the following facts are to be noted :—

When one ampere of current is impelled by a pressure of one volt, the *rate of dissipation of energy* is called one "watt." *Note that a watt is a rate of doing work*, or a rate of dissipating energy. In the same way a "horse power" is a *rate* of doing work, and means such a rate that 33,000 foot-pounds of work is done *in one minute*. It is not an absolute quantity of work, but a *rate* of doing work. Just so, a "watt" does not correspond to a certain number of thermal units, but to a certain number of thermal units *per unit of time*.

If one watt flows through a resistance for one minute it produces exactly the same amount of heat as the dissipation of 44 foot-pounds per minute, so that there are

33,000/44 = 74 6 watts in one horse power.

One watt acting for one hour therefore obviously produces

44 × 60 = 2,640 ft. pounds = 2,640/778 = 3.4 B.T.U.

1 watt = 3.4 B.T.U. per hour.

If, therefore, we have a current of, say, 25 amperes at 220 volts, the amount of heat which will be produced by that current in one hour is

25 × 220 × 3.4 = 18,600 B.T.U. per hour.

Again, if we require to produce heat by means of an electrical current at the rate of, say, 8,000 B.T.U. per hour, we should require

8,000/3.4 = 2,350 watts.

If the voltage is 220, the amperes required would be

2,350/220 = 10.7

The maintenance of 1,000 watts for one hour is the Board of Trade *unit of electrical quantity.* One Board of Trade unit is obviously therefore the equivalent of

1,000 × 3.4 = 3,400 B.T.U.

If, therefore, we are buying electrical energy at the rate of 1½d. "per unit," in order to supply, say, 8,000 B.T.U. per hour, we should require

8,000/3.4 = 2,350 watts

and

2,350/1,000 = 2.35 units per hour

would cost

2.35 × 1.5 = 3½d. an hour.

The heat developed per minute by any current is given by the formula $E\ C/17.7$, or $C^2\ R/17.7$. These are the figures that are chiefly employed by calculations which the heating engineer will require.

The approximate amounts of electrical energy consumed in various sizes of new carbon filament lamps are given by the makers as follows :—

Candle power.	Volts.		Amperes.	Watts.	Watts per candle power.
1	3	×	0.8	2.4	2.4
2½	5	×	1¼	6.25	2.5
5	5	×	3	15	3
8	10	×	2.8	28	3.5
16	15	×	3.7	55.5	3.46
25	40	×	2.2	88	3.52
32	50	×	2.3	115	3.6
50	50	×	3.5	165	3.5
100	50	×	.7	350	3.5

The heat developed in a lamp is given as above by the formula W = E.C.T./17.7 where T is the time in minutes. (9)

The current consumed by the metallic filament lamps is very much less than these, as the following figures will show :—

Volts.	Watts.	Candle power.	Watts per candle power.
50	13	10.5	1.24
50	17	14.5	1.17
100	25	22.5	1.11
135	32	28.5	1.12
200	55	45	1.22

The mechanical power developed by a motor in horse power may generally be taken to be from 50% to 85% of the power represented by the product of the volts and amperes, the remaining 50 to 15% being wasted in producing heat or noise, or other manifestations of energy.

$$\text{Horse-power} = E \times C/746 \times 85/100$$

for a motor of high efficiency, or if the required horse-power is given, the watts necessary to produce the power may be found thus for a motor of low efficiency :—

$$\text{Watts required} = \text{horse-power} \times 746 \times 100/50.$$

On Angular Velocities.

If any body is rotated about an axis it is said to have an "angular velocity." The magnitude of the angular velocity is measured by the number of units of angle described per second by any line in the body perpendicular to the axis of rotation. The angles for this purpose are measured in radians, one radian being an angle the length of whose arc is equal to the radius.

There are clearly π of such angles in two right angles (180°), therefore

$$1 \text{ radian} = 180° \div \pi = 57.4° \text{ about.}$$

Centrifugal Force.

Any body of mass m pounds which is at any instant moving round a curve of radius r feet, with a velocity v feet per second, experiences a force acting outwards from the centre of the curve

$$= \frac{m\,v^2}{r} \text{ poundals.}$$

If ω be the "angular velocity," then

$$\omega\, r = v$$

so that

$$\text{Centrifugal force} = \frac{m\,v^2}{r} = m\,\omega^2\,r \qquad (10)$$

This equation gives the force in poundals. If the force is required in pounds weight, the value is

$$\frac{m\,\omega^2\,r}{g} \quad \text{or} \quad \frac{m\,v^2}{r\,g}$$

A more complete explanation of this will be found in any good book on mechanics, or in the Author's "Graphic Methods of Engine Design," pp. 96-101.

The Laws of Hydrostatics.

So far as the heating engineer has to deal with the laws of hydrostatics, the following are the leading principles with which he must be familiar.

Formal proofs of the various theorems may be obtained from any good book on hydrostatics. Space cannot be spared here.

Definitions.

The *density* of any substance, solid, liquid, or gaseous, is the number of pounds mass in one cubic foot of the substance.

The *specific gravity* of any substance is the ratio of its density to that of water at standard temperature and pressure, which is generally taken as 62.4 lbs. per cubic foot. A more exact value is 62.355.

The Pressure produced by any Liquid.

Assume a vertical tank, one square foot in cross section, is filled with liquid, whose density is D pounds per cubic foot to the height H feet. The total pressure on the base of such a tank is obviously the weight of H cubic feet of liquid, each of which weighs D pounds. The total force on the base of the tank is therefore $H\,D$ pounds. On every square foot at the same depth the pressure is the same. The pressure at any point x feet below the surface is in the same way $x\,D$ pounds per square foot.

This pressure acts in all directions equally, which is a property of the phenomenon known as a "fluid pressure." That it acts upwards in a mass of fluid as well as downwards will be understood by considering a thin layer of the liquid, say, .01 foot thick, at the very bottom of the tank. The weight of liquid above this layer must be supported by *something*. It is only in contact with the thin layer in question, therefore the upward force which the thin layer must exercise on the column of liquid above it must be equal to

$$D \times (H - .01).$$

The pressure also acts at right angles to the vertical direction, or in any other selected direction with an equal intensity. Proof of this can be obtained in any book on hydrostatics.

If a hole is drilled into the tank just above the bottom, it is well known from experience that it will take just as much force to stop up this hole as it would if the hole were in the bottom of the tank.

If the tank is filled with three or four or more different columns of liquid of heights H_1 H_2 H_3, and of different densities D_1 D_2 D_3, one over the top of the other, the pressure on the bottom of the tank is the sum of the weights of all these liquid columns or

$$\text{Total pressure} = H_1 D_1 + H_2 D_2 + H_3 D_3 + \text{etc.} \qquad (11)$$

If the heights are given in feet and the densities in pounds per cubic foot, the pressures given by the formula are in pounds per square foot.

If two columns of liquid of different densities are balanced against one another in a U-tube, the pressure due to each column must be the same, otherwise the column generating the greater pressure would force the other liquid back up its own limb of the U-tube.

In this case

$$H_1 D_1 = H_2 D_2$$

If a solid body is immersed in a mass of liquid, it follows from the principle of the conservation of energy that there must be an upward force acting on the solid body equal to the weight of the liquid displaced, that is to say, the solid body weighs less in the liquid than it would in vacuum, by the weight of an equal volume of the liquid.

The reason for the existence of this upward force will be understood by imagining that the immersed body is a vertical cylinder of weight w lbs., of height h feet, and cross section s square feet. Its volume is $h\ s$ cubic feet. The top face is H feet below the surface. Then

Total pressure on top face due to liquid $= H\ D\ s$ pounds downwards.

Total pressure on bottom face $(H + h)\ D\ s$ upwards.

Nett upward pressure $(H + h)\ D\ s - H\ D\ s.$

$$= h\ s \times D.$$

= weight of a volume of liquid equal to the volume of the solid.

If the substance of the solid weighs more foot^3 for foot^3 than the liquid the body will sink, because gravity pulls it down more powerfully than the liquid pushes it up.

If the solid is lighter foot^3 for foot^3 than the liquid it will float up to the surface, because the liquid pushes it up more powerfully than gravity pulls it down.

This forms a convenient method of ascertaining the volume of any body of regular shape, or its density, or the density of a liquid.

Units.

The question of the units in which any formula is worked out is one of the greatest possible importance. British speaking countries are afflicted with a double system of units, one of which, known as the centimetre-gramme-second, or the "C.G.S." system, is almost universally used by the scientific man who evolves the fundamental theory on which all engineering science is based. The other, known as the foot-pound-second, or F.P.S. system, is that in current use in the commercial world.

It is not easy to speak too severely of the disadvantage which this double system of units entails on the English speaking races, nor of the ridiculous character of many of the British units, the use of which in commercial life should be abolished by Act of Parliament.

The results of all the work done in foreign countries are expressed in the C.G.S. system of units, and if the British engineer is to make use of some of that most valuable work, he must either learn to think in C.G.S. units, which is an extremely difficult thing for one who is accustomed to work in foot-pound-second units, or he must learn to convert the results and formulæ from the one system to the other, which is an operation involving considerable difficulty and a great deal of care.

It is, indeed, very difficult for any person who has not had a training in mathematical science to grasp the complicated inter-relations involved in the conversion of formulæ from one system of units to another. A close adherence to the principles enunciated below will, however, enable the actual process of conversion to be accomplished without error, even though the exact principles involved are not fully grasped.

Dimensions of Units.

Every physical unit of every kind may be expressed in terms of the three fundamental units of mass, length, and time. The expression which shows what is the relation of any particular physical unit to the three fundamental units is called the "equation of dimensions," and a thorough understanding of the theory of the equation of dimensions is a very great safeguard in the application of arithmetic to formulæ in enabling mistakes to be detected.

Simple Units.

The three absolute units are indicated respectively by l, t, m.

The unit of area, the square foot, is $l \times l = l^2$.

Unit of cubic capacity $= l \times l \times l = l^3$.

The unit of velocity is a certain distance per unit of time, it has therefore the dimensions $\frac{l}{t}$

The unit of acceleration being an increase of a unit of velocity per unit of time $= \frac{l}{t} \div t = \frac{l}{t^2}$

Force being mass × acceleration $= \frac{m\,l}{t^2}$

Energy = force × distance $= \frac{m\,l^2}{t^2}$

Pressure being a force per unit area $= \frac{m\,l}{t^2} \div l^2 = \frac{m}{t^2\,l}$ similarly with all other units.

Compound units are more difficult to treat in this exact manner on account of the fact that the "dimensions" of their components are often by no means clear. For instance, a rate of transmission of heat through a surface has as dimensions a number of heat units per unit of area per unit of time, per unit difference of temperature.

The "dimensions" of a heat unit are not at once clear. In order to obtain them we have to note that heat is a form of energy. Energy is force × distance.

The dimensions of a unit of heat are therefore $\frac{m\,l^2}{t^2}$

Although this may be perfectly accurate, the meaning of it is not immediately comprehensible. The "dimensions" of a unit difference of temperature are still more involved.

Although the theory of dimensions is perfectly applicable to such compound units, it is better to treat them independently of the theory of dimensions, because of the difficulty of understanding their meaning.

The following schedule of such units shows how they can be transformed from one system to the other, and a table of the particular compound units with which the heating engineer will be chiefly concerned is given in the Appendix.

Formulæ.

Any physical equation expressing equality between two quantities must be such that the dimensions of the units in which each term of the equation is expressed must be the same. If they are not, the equation is incorrect.

The relation between two equations representing the same phenomenon in different units can be ascertained by merely determining the "dimensions" of each term, and altering each of the fundamental units in proportion to the size of the unit in each case.

This is a perfectly accurate method of determining this relation, but it is too difficult to understand to be generally used.

An easy and straightforward process of converting correctly a formula expressed in C.G.S. units into one expressing the same result in British or any other units is as follows :—

In order to avoid confusion of thought it is desirable to leave out of count the dimensions of the formula altogether and to consider the equation merely as an arithmetical sum, which expresses the identity of result of two arithmetical operations represented by the two sides of the equation.

First convert each symbol of the original expression to the *same numerical value,* expressed in terms of the new units.

The new equation so produced will thus be numerically identical with the old one. If the old one is accurate the new one must also be accurate. Then combine the numerical co-efficients used in effecting the transformation into one figure. This will be the "conversion factor."

Care must be taken that all the units involved are expressed in the equation and not merely implied. Take a simple case.

Let

$$D = A\ VT$$

where D expresses the distance travelled by a man in kilometres in time T minutes, where V is his velocity in metres per second, and A is a co-efficient suitable for these units.

The value of A is here obviously

$$\frac{60}{1{,}000} = .06$$

The formula therefore becomes

$$D = .06\ V\ T$$

To express this formula in miles, feet per second, and hours, we proceed as follows :—

1 mile = 1.61 kilometres.
1 metre per second = 3.28 feet per second.
1 minute = $^1/_{60}$ hour.

The original equation may be written

$$(\text{Miles} \times 1.61) = A \times \frac{v \text{ feet per sec.}}{3.28} \times (t \text{ hours} \times 60).$$

This is precisely the same numerical sum as the original equation where A has the same value in each case.

Combining the co-efficient

$$\text{Miles} = A \times (v \text{ ft. per sec.}) \times t \text{ hours} \times \frac{60}{1.61 \times 3.28}$$

$$\text{Miles} = A\,vt \times 11.4.$$

$$\text{Miles} = A\,vt \times 11.4 \text{ or Miles} = (11.4\,A)\,vt$$

$$\text{or Miles} = (11.4 \times .06)\,vt = .685\,vt$$

The original formula $D = A\,VT$ in the original units therefore is transformed into a new formula

$$d = a\,vt$$

where the value of a is 11.4 times the value of A.

Every equation used by an engineer must essentially bear a similar numerical interpretation.

Consider, for instance, the equation

$$W = \frac{K\,(T_1 - T_2)\,A\,H}{E}$$

giving the number of calories transmitted in H hours through A square metres of a plate of any material, say, iron, whose conductivity is K and whose thickness is E metres, when a temperature T_1° C. is maintained in the material on the one side, and T_2° C. on the other.

This equation is essentially numerical in meaning, it states that if the heat passing is expressed in calories, the difference of temperature in degrees C., the area in square metres, and the thickness in metres, then the number of calories per hour =

$$K \times \text{difference of temperature} \times \text{area} \div \text{thickness}.$$

The values of K for different materials are given in tables suitable to these units.

Assume that it is required to transform this equation into another which will give the result in B.T.U. when the temperatures are expressed in degrees Fahrenheit, the area in square feet, and the thickness in feet.

Now there are 3.968 B.T.U. in one calorie.

1.8° F. in 1° C.

10.76 square feet in one square metre.

3.28 feet in one metre.

If, therefore, the original equation is numerically correct, it can be written in this form.

$$\left(\frac{\text{B.T.U.}}{3.968}\right) = \frac{K\left(\dfrac{t_1\,°\text{F.} - t_2\,°\text{F.}}{1.8}\right)\left(\dfrac{\text{sq. ft.}}{10.76}\right) \times \text{H}}{\left(\dfrac{\text{thickness (feet)}}{3.28}\right)}$$

where K has the same value as before. Note that *this is precisely the same numerical equation as before,* each numerical term in brackets is identical with the corresponding term of the original equation. The co-efficients in this last equation can now be collected as follows :—

$$\text{B.T.U.} = \frac{K\,(t_1\,^\circ\text{F.} - t_2\,^\circ\text{F.}) \times \text{sq. ft.} \times \text{H}}{\text{thickness (feet)}} \times \frac{3.968 \times 3.28}{10.76 \times 1.8}$$

$$= \frac{K\,(t_1 - t_2) \times a \times H}{e} \times 0.672$$

This equation will obviously hold good when *K* has the same value as for C.G.S. units.

If, therefore, every value of *K* given in the table is multiplied by 0.672 a new set of values for *K* will be obtained which will make the original equation hold good for British units. Any equation in the C.G.S. units can be correctly transformed into any other units by the adoption of this device, even though the real mathematical nature of the operation is not fully comprehended. The essence of the process is to write the equation in such a form that the numerical value of each term in it shall be the same as in the original equation, afterwards combining the co-efficients.

CHAPTER II.

PROPERTIES OF HEAT.

In the last chapter heat was stated to be in reality a form of energy. The meaning of this statement is that a definite quantity of mechanical or other energy can always be converted into a definite quantity of heat without any by-products.

The process can also be reversed, a definite quantity of heat can be converted into a definite quantity of mechanical work with no by-products, except heat and work.

Owing to practical difficulties in manipulation, though there is no actual destruction or diminution of quantity, the loss or *escape* of energy during a continuous process of transformation from a lower to a higher form is often or always relatively great.

Heat is that form of energy into which all other forms have a tendency to degenerate. Given a fair chance, almost all other forms of energy will automatically turn into the form of heat. It may be described as the lowest, least specialised, or most degenerate, unorganised, or amorphous form of energy. It probably consists of kinetic energy of individual molecules of the heated substance. Each molecule is supposed to have a rapid vibratory movement. The sum of the vibratory energy of all the molecules is regarded as the heat energy.

By certain means the energy of individual molecules of a hot body can be collected and converted into mechanical or other forms of energy, with a corresponding diminution in the molecular or heat energy of the body.

Fundamentally, therefore, heat and mechanical energy must be one and the same thing in a different shape.

Heat is invisible and impalpable. Its existence can only be indirectly recognised by the observation of its effects. One can feel the conduction of heat into one's own body by touching a hot object. One can see heat melting ice or iron, expanding mercury, or air, and in other ways. Generally, but not always, the passage of heat into or out of a body can be recognised by an alteration of the " temperature " of the body.

The meaning of the expression "temperature of the body" can be partly, but only partly, appreciated by the sense of touch, *e.g.*, by placing the hand on the body. Its real meaning can only be grasped by considering some of the fundamental properties of heat.

The most characteristic property of heat is its tendency to communicate itself from one body to other neighbouring or touching bodies.

If two bodies are placed side by side there will always be an interchange of heat between them, whatever their relative temperatures. If they differ in temperature, the body at the higher temperature will communicate its heat more rapidly than that at the lower temperature. The interchange of heat between the two will therefore result in the body at the lower temperature receiving heat at a greater rate than it parts with it, and thereby it gains heat.

The hotter body, on the other hand, parts with more heat than it receives from the cooler body. The general effect of the interchange, therefore, is that the hotter body loses and the cooler body gains heat.

Two bodies at different temperatures cannot indeed exist side by side without producing this result.

It is a manifestation of an essential property of heat.

The "temperature," in fact, is a measure of the tendency which heat has so to diffuse itself, to expand or to communicate itself to surrounding bodies, just as the pressure at which a pound weight of air is compressed into a cylinder is a measure of the force with which the air will expand if it has a chance.

The quantity of heat in a body is thus an essentially different thing from the temperature of the body, just as a quantity of air is a different thing from its pressure, or a quantity of electricity from its voltage.

Thus heat has different grades, the grade being represented by the temperature. When at a higher temperature it has a tendency to become degraded into the same quantity of heat at a lower temperature. It is this property of heat with which the heating engineer has chiefly to deal.

Units of Heat and Temperature.

It is easier to give an intellectually satisfactory definition of a unit quantity of heat than of a degree of temperature, without dealing with somewhat abstruse scientific considerations.

It has been explained above that one unit of potential or mechanical energy can always be converted into an absolutely identical equivalent in heat by the mere process of allowing the unit of mechanical energy to expend itself solely in producing friction. Some multiple of the unit of mechanical energy (the foot-pound) can therefore be used as the unit of heat. Thus, suppose a mass of one pound is placed in such a position that it can fall through one foot, and is coupled up to gearing or other mechanism of any kind in such a way that it slowly falls through one foot, the energy thus liberated being entirely consumed in producing either friction in the mechanism or some artificially arranged friction. It has been ascertained by careful experiment that in these circumstances a certain definite and invariable quantity of heat is always generated.

It does not matter whether the temperature of the bodies in which heat is produced is high or low, exactly the same quantity of heat will be produced.

This definition of a unit of heat, as will be seen, does not involve any postulates with regard to the properties of bodies. It merely assumes the substantial identity of different kinds of energy and the fundamental law of nature, known as the law of the "conservation of energy," which we must regard as a postulate. Such a definition is intellectually satisfying.

Similarly we could always be sure of producing a definite quantity of heat irrespective of its temperature, by allowing a certain definite electrical current to pass through a definite resistance for a certain time. This method involves more complicated calculations, and the use of more complicated ideas, instruments and units than the other imaginary method of generating one unit of heat.

Other similar methods of generating a definite quantity of heat might be devised which would be absolute and satisfactory within certain limits.

Unit Difference of Temperature.

It is by no means, however, so easy to describe one unit difference of temperature without a great deal of scientific explanation. We can always, by our sense of touch or by some easily imagined means, theoretically place any number of bodies, numbered, say, 1, 2, 3, 4, 5, etc., which are at different temperatures, in the order of their temperature, in such a way that each body is at a higher temperature, say, than the body next to it on the right.

But the question now to be discussed is what is meant when we say there is the same, or a less or a greater difference between bodies Nos. 2 and 3 than there is between bodies Nos. 4 and 5?

The only property of temperature to which reference has yet been made is that a body at a higher temperature when placed in contact with another body at a lower temperature will, as the nett result of a mutual transference of heat, communicate some of its heat to the body at the lower temperature, and will continue to do so until the temperatures are exactly the same.

The obvious suggestion is then that the difference in temperature between two bodies, 2 and 3, should be defined as the same as the difference between two other bodies 4 and 5, if when placed in contact respectively and in the same conditions 2 communicates the same amount of heat per second to 3 as 4 does to 5.

Reflecting on such a definition as this, however, the question at once arises, how do we know that the conductivity between 2 and 3 at the higher temperature is the same as that between 4 and 5 at a lower temperature? Does the absolute difference in temperature between the two systems make any difference in the rate at which heat is transferred?

In the attempt to make such a definition valid and convincing we are compelled to postulate that 2 3 4 5 are made of the same material, and that both they and the conditions in which they are placed have the same properties as regards conductivity at the upper as at the lower temperature. We are driven to find a definition of conductivity, which necessarily in its very nature involves expressions relating, among other units, to that of temperature. On this method of definition we are arguing or defining as it were, in a circle.

Or since we are now provided with a provisional unit quantity of heat equivalent differences of temperature in different parts of the scale might be defined as such respective rises in temperature as are due to the communication of one unit of heat to a unit quantity of some definite substance at different temperatures.

But here the question arises how do we know that the same amount of heat *will* raise one unit of any particular substance through what is *really* the same range of temperature in all parts of the scale? That is to say, how do we know that it is logical or scientifically sound to assume this? As a fact we know that in the great majority of cases one unit of heat produces different rises of temperature in different parts of the scale.

In order to make it sound we should have to postulate a substance with a constant "specific heat." But we have not yet defined the meaning of specific heat, and in order to do so we should have to introduce into our definition the unit of temperature. So that this attempt at a definition also in reality only restates the problem in another form.

Or we may again for our definition rely on the well-known fact that the same body at a low temperature alters in dimensions when raised to a higher temperature. We might take some standard substance, say a bar of iron, and arbitrarily fix 1 deg. difference of temperature to be such a rise in temperature as will increase the length of a bar of iron by one hundred thousandth of its length. This would be similarly unsatisfactory, because we do not know that what is in reality the *same* rise in temperature would produce a corresponding increase of length at different temperatures. Indeed, as a fact, we know that it does not, and that different bodies in various ways do not only not agree among themselves as regards expansibility by heat, but that any one body does not in fact expand with exact uniformity.

It is possible to conceive other properties appertaining to differences of temperature which might be utilised for the purpose of defining what we mean by equivalent intervals of temperature, all of which conceivable definitions would be equally unsatisfactory and for a similar reason.

Which of these definitions is to be accepted, and why should one be accepted rather than the other? In fact, what is the fundamental conception of one degree difference of temperature?

In order to find a definition which is wholly satisfying to the intelligence, it would be necessary to go deeper into the science of thermodynamics than it is proposed to do in the present book. Those who wish to understand the question thoroughly may read Clarke Maxwell's classical work on "The Theory of Heat." It will be here sufficient to say that this scientific or intellectually satisfying scale of temperature is fundamentally based on the conditions in which heat is convertible into mechanical energy, and on the conception of an ideally perfect heat engine of Carnot, the French scientist.

Lord Kelvin showed that this beautiful conception could be employed to give a rational basis for a scale of temperature which should be as independent of the properties of any single substance as is the unit of heat.

This scientific scale of temperature is found in practice to be so close to the scale now about to be explained, that the student of heating engineering need not trouble himself about the difference, although he should take note of the fact that a true scale of temperatures is only founded on the expansion of any particular substance for convenience sake, and not because it is scientifically satisfactory.

It is found experimentally that for all practical purposes the definition now about to be explained, may be accepted as a satisfactory standard. The student must take it on trust that there is sound reason for regarding the difference of temperature between say 194° and 195° F. as *the same difference of temperature* as exists between say 44° F. and 45° F., independently of the fact that mercury or air happens to expand with moderate uniformity for real equal rises of temperature.

Practical Units of Temperature and Quantities of Heat.

There are certain natural phenomena known to us which fix certain definite points on the thermometric scale, altogether irrespective of our theorising about equivalent intervals. As these phenomena are easily reproduced, and are of constant occurrence in nature, it is found to be not only the most convenient but the most scientific method to start all practical calculations on thermometric scales, with these two points as basis.

It is found that whatever criterion of the temperature is adopted, the temperature at which pure ice melts, at a constant pressure, is always the same, and that at a constant and standard pressure the temperature of steam from boiling water is always the same. There is universal agreement among scientific men that these two points shall form the fixed points of all practical thermometric scales.

Water is a substance which is easy to obtain in a state of purity, and the two temperatures at which it freezes and boils respectively at atmospheric pressure are well within the practical range. Further, almost all the temperatures with which we have to deal in practice are also either within that range or within quite a measurable distance of it.

The problem therefore reduces itself to that of sub-dividing this range of temperatures accurately, so that one may say that the difference represented by any number of degrees in the lower part of the scale is the same difference as that of the same number of degrees in any other part of the scale.

The numerical measure of temperature is found by utilising the property of heat, whereby all bodies alter in their dimensions when the temperature is changed. A body at the freezing point of water is always different in volume and length and breadth from the same body when at, say, boiling point. Further, it is found that any solid body has always the same dimensions at the same temperature and pressure, unless the process of heating causes some molecular or chemical change in the nature of the body. A bar of iron which at freezing point is exactly 36″ long, at boiling point is always, say, 36.0438″ long. If the temperature is again reduced to freezing point the length again becomes 36″.

If, therefore, we are provided with such a bar of iron which we know is 36″ long exactly at freezing point, and if at the same time we have a micrometer which enables us to measure its length very exactly, we can at any time reproduce any desired "temperature" by warming this bar of iron until it has a particular length which is known to correspond to the temperature to be reproduced.

It would be possible to use any known substance for a thermometer if its properties had been investigated with sufficient care. But different substances do not expand regularly between different temperatures. Thus, three bodies, A B C might all have the same length, say, 100 inches at freezing point. If they were all heated up to some common temperature between freezing and boiling points, A might be 100.2″ long, B 100.21″, and C 100.22″. If they were then heated up to boiling point their length might be as follows :—

A 100.3″, B 100.26″, and C 100.28″.

We might, of course, agree to take any one of these substances as the standard, and say that one degree difference of temperature was such a difference as would make the bar of this particular material increase in length by such and such a fraction of its total expansion between freezing and boiling points. It will easily be understood that since substances differ between themselves, such a method would be absolutely arbitrary. It would depend upon the properties of one particular substance. It is a fundamental scientific principle that physical units should be defined independently of the properties of any one particular substance. A very serious drawback of such a method would be that a bar of any known material expands so little that a very long bar would be necessary to give indications which could be read off without using a micrometer, so that a solid in any case would be unsuitable for a thermometric substance.

It is found that all *gases* when far from their condensing points have substantially the same co-efficient of expansion, and not only so, but their indications at any particular temperature agree, and therefore that a *gas* thermometer always has approximately the same reading *whatever the nature of the gas employed.*

Although this fact does not prove that a gas thermometer would give correct indications of temperature, yet for present purposes we may take the gas or air thermometer as being that which is *most likely* to give correct intervals. The indications of a properly calibrated gas thermometer are in fact very close indeed to the indications of a truly scientific scale. They are not absolutely correct, because no known gas is an absolutely "perfect gas." A "perfect gas" would be one which obeyed exactly the laws of Boyle and Charles, as explained in the chapter on properties of air. If nature provided what is called an absolutely "perfect gas" its indications would be absolutely correct and absolutely the same as a truly scientific scale of temperature. This is indeed the criterion of a perfect gas, although, as stated above, a true scale of temperature is not strictly based on expansion at all.

In practice, however, a gas thermometer is awkward to use as a practical instrument except for special purposes. It is bulky and its indications vary with the pressure and humidity of the air. It is found in practice that by far the most convenient device is a liquid thermometer, which can be sealed up out of contact with the air, so that alterations in the atmospheric pressure do not affect its readings, and its volume in any case varies very little with the pressure. Further, the differences in the indications of a suitable liquid and a perfectly correct gas thermometer are not sufficiently great to cause any practical difficulty.

A liquid thermometer is in fact a very long thin bar of liquid of which the greater part has been rolled up into a bunch confined into a bulb, so that only the end of the liquid rod is visible. The expansions and contractions of this end show how much the very long bar of liquid has expanded and thus give the indications of temperature.

For practical purposes mercury is the best material to employ for a thermometric substance, because it has a fairly high co-efficient of expansion and because it does not easily vaporize. It is easily visible in a glass tube, it also has a certain amount of cohesion with its own substance and does not cling to glass. Suitably treated alcohol and other liquids are also sometimes employed.

The practical division of the thermometric scale is effected by first marking on the scale by trial the freezing and boiling points, labelling these points for the Fahrenheit scale 32° and 212° respectively, and subsequently sub-dividing the expansion between these temperatures into 180 equal parts, care, of course, being taken that the bore of the tube is uniform. On the Centigrade or Celsius scale the points are labelled 0° and 100° respectively, and the interval sub-divided into 100 equal parts.

The indications of a thermometer thus sub-divided are for practical purposes the same as would be the indications of a gas thermometer similarly sub-divided, and also practically the same as a truly scientific scale of temperature. The greatest difference is about 0.01 degrees. This scale of temperature may be taken to be a satisfactory scale in such a sense as the following.

If we may postulate a substance which has a constant specific heat (which we may assume independently determined) between the temperatures of 32° F. and 212° F., then it would take the same quantity of heat (in other words it would absorb the same quantity of work or potential energy) to raise a given weight of that substance through the same number of degrees in any part of this mercury scale. Thus, the amount of work which it would be necessary to expend in order to raise the temperature of 1 pound of water (assuming its specific heat to be uniform, which it is nearly, though not absolutely*) from say 130° to 135° would be 5 × 778 foot-pounds (see below for explanation of this). An exactly equal number of foot-pounds would be required to raise the temperature of the same quantity of water from 40° to 45°, or from 95° to 100°, or from 180° to 185°, as measured by our thermometer. This for the present may be taken to be an illustration of the meaning of the statement that the intervals on an ordinary well made thermometer are rationally and accurately sub-divided.

Fahrenheit and Centigrade Scales.

There are only two different scales of temperature with which the heating engineer is concerned, the Fahrenheit scale and the Celsius or Centigrade scale.

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* The specific heat of water at the boiling point is about 5% greater than at freezing point. (See Chapter IV.).

(1) In the former the freezing point of water is called 32°—owing to the mistaken notion of the originator of the scale that 0° F. represented the lowest attainable temperature. For some equally mistaken reason boiling point is called 212°, so that the interval between freezing and boiling points is divided into 180 equal parts.

(2) The Celsius or Centigrade scale, in which the freezing point is called 0° and the boiling point of water 100°.

Comparing these two scales, 1° C. is equal to 180/100 = 1.8° F., or 1° F. is equal to 5/9ths of 1° C.

The process of converting a temperature from one scale to the other is as follows :—

(1) *From Fahrenheit to Centigrade.*—Ascertain the number of degrees *above freezing point* on the Fahrenheit scale and multiply by 5/9. This gives the temperature on the Centigrade scale.

(2) *From Centigrade to Fahrenheit.*—The reverse process is, multiply the temperature on the Centigrade scale by 9/5ths and add 32°.

The tables given in the Appendix give the corresponding values of each degree between boiling and freezing point.

Quantities of Heat.

A strictly scientific measurement of a quantity of heat would be the amount of heat produced by the degradation of a certain number of units of work. If we were to take one foot-poundal of work as the unit, the amount of heat which would be brought into existence by the destruction by friction or otherwise of one foot-poundal of work would be one unit of heat. Such an unit would have the advantage that it is independent of the unit of temperature, and would be an absolute unit in the strictest sense.

As a practical unit, however, it would be far too small. Not only is it too small, but such a definition of a heat unit would be extremely inconvenient, because of the practical difficulty of making rough experiments to ascertain quantities of heat by a direct process. Further, such a description would be completely out of touch with the practical use of heat units with which we are chiefly concerned.

For all these reasons a much more convenient unit of heat is found to be that which will raise one pound of water through 1° F. (*i.e.*, from 39° to 40° F.), and this in English speaking countries is universally accepted as the practical unit of heat or "British Thermal Unit" (B.T.U.).

In Continental countries, and countries adopting the C.G.S. system of units, the quantity of heat generally accepted as a practical unit is the "calorie," which is the amount of heat necessary to raise 1 kg. of water from 0° C. to 1° C. This unit is about 3.968, or roughly four times as large as our English unit.

Such units as these do not, of course, fulfil pedantic scientific ideals. They are not independent of the properties of any particular substance. This fact does not cause any inconvenience either in scientific or practical work. Heating engineers deal with quantities of heat in large masses. Their calculations are in their very nature approximate only. It is sufficient for a heating engineer to know that one B.T.U. (or 778 foot-pounds of work degenerated by friction into heat) will raise 1 pound of water through about 1° in any part of the scale, and that a unit of heat obtained by cooling 1 pound of water from, let us say, 181° to 180° F. is approximately equivalent in effect to a unit of heat obtained by cooling 1 pound of water from, say, 41° to 40°. The fundamental difference between these two is that the unit of heat at 181° can be communicated by mere contact to any other body at any temperature below 181°. It could be used for instance for heating another pound of water from 150° to 151°, whereas the unit of heat at 41° could only be used for heating, by simple contact, bodies at a temperature below 41°.

It is, however, possible to employ heat derived from a body at a relatively low temperature, in order further to heat bodies at a higher temperature, by employing a certain amount of mechanical work in order to raise that heat to a higher temperature. Heat at a lower temperature can be raised to a higher temperature by means of mechanical work, with an increase in its own quantity corresponding to the amount of work expended on the process. This is a thermo-dynamic matter, with which heating engineers, at the present stage of heating engineering, have nothing to do. This principle will probably become of great importance in the distant future.

If we wish to heat air at, let us say, 60°, any heat which is already degenerated to 60° or below is absolutely lost for the purposes of the heating engineer. It is, therefore, important at every stage to maintain heat at a temperature higher than that at which the heat is to be used, and not to allow it to degenerate so much that it cannot be readily transferred, for although by degenerating in temperature it does not lose any of its quantity as heat, yet it becomes less and less available as its temperature becomes lower and lower.

Since, however, as has been pointed out, the higher the temperature at which heat exists the greater is its tendency to escape, the heating engineer must see to it that when he wishes to transfer heated substance to a distance or to prevent heat from escaping, its temperature must not be higher than is necessary for the purpose in hand. It is economical to maintain heat at as high a temperature as possible when it has no chance of escaping or when *he wishes* it to escape rapidly in any desired direction. But when he does not wish it to escape it must be maintained at as low a temperature as is consistent with the efficient performance of the work in hand.

Specific Heat.

The specific heat is a measure of the capacity a body has for absorbing heat. It may be defined as the quantity of heat which has to be added to unit mass of a body in order to raise it through unit difference of temperature.

Bodies differ greatly in this property, thus 1 pound of water will absorb about nine times as much heat as a similar weight of iron when heated between the same limits of temperature. A table of the specific heats of the various substances with which the heating engineer is concerned is appended.

This is one of the most valuable properties which water possesses, namely, its high specific heat relatively to other bodies. The heating engineer therefore uses water very largely as a heat carrying medium. It will carry a relatively large body of heat without being unduly raised in temperature. It also has for a liquid a remarkably high specific gravity, so that a considerable mass of water only occupies relatively little space.

On Absolute Temperature.

Experiments on the heating and expansion of gases, especially of air, have led physicists to the conception of a very low temperature, at which all the heat or molecular energy existing in a body has been drained out of it.

It is not possible to assert definitely that at what is called the "absolute zero" of temperature there is absolutely no heat in a body, because that temperature has never yet been attained and probably never will. Little definite experimental knowledge of the subject exists. It is assumed, however, that at the temperature which is called the "absolute zero" all molecular energy is absent.

The conception has been arrived at in the following way:—

It is found that when a mass of air at 0° F. is cooled 1° it reduces in volume by about 1/461 of its volume at 0° F. if the pressure remains constant, or by 1/461 of its pressure if its volume remains constant. A further drop in temperature of the same amount results in a further reduction *of the same absolute amount.* Still further reductions of temperature produce reductions of volume or pressure *always of the same absolute amount per degree.* If air obeyed the same law right away down to a very much lower temperature than has ever yet been attained, it is clear that its pressure would have disappeared altogether at a temperature about 493° F. below the freezing point of water. What does actually happen at a very low temperature of this kind is that the gas condenses into a liquid when the laws connecting its pressure and volume become irregular.

But it is assumed, for want of experimental verification, that the fact that air and all gases behave in an identical manner when sufficiently far removed from their condensing point shows that at or about a temperature of 461° F. a perfect gas would lose all of this molecular energy, and this temperature is therefore called the "absolute zero" of temperature.

The value of this conception to the engineer or physicist is that by its aid it is easy to calculate problems involving the volume and pressure of gases at different temperatures. The student need not trouble himself about the exact meaning of the expression "absolute scale of temperatures," but use it merely as a convenient expression of the law of expansion of air or any other normal gas and take the absolute zero to mean the bottom of the scale of the gas thermometer. Further information on this point is found in the chapter on Properties of Air.

CHAPTER III.

PROPERTIES OF AIR.

THE atmosphere in a pure state consists of a mixture of the gases oxygen and nitrogen, in the proportion of 21 parts by volume of oxygen, and 79 parts of nitrogen; by weight, 24 parts of oxygen and 76 parts of nitrogen. The normal outside air also contains about 3.5 to 4 parts in 10,000 of carbon-di-oxide, a varying proportion of water vapour (as to which see the section devoted to that question), and a very small proportion of other gases with which the heating engineer need not trouble himself.

The standard pressure of the atmosphere is 14.7 pounds per square inch, or 1.0335 kilogrammes per square centimetre = 29.922, or nearly 30 inches of mercury, or 33.947 feet of water column at the sea level. This is the standard pressure for which all calculations in ventilation are reckoned.

At heights above the sea level the pressure of the air falls according to a definite law. For practical purposes the values in the table may be taken to be the pressures in inches of mercury and in pounds weight per square inch at different heights above the surface of the sea.

At these heights the air becomes more rarefied, as will be seen by reference to the later parts of the present chapter.

Expansion of Air due to Alterations of Temperature.

In common with all other gases air has the property of a high coefficient of expansion. When the pressure is maintained constant the volume of any mass of air is proportional to its absolute temperature (see page 27), *i.e.*, where t° is the temperature F., the volume is proportional to

$$461 + t^\circ.$$

The problems relating to the expansion of air, due to changes of temperature alone will be best understood by consideration of the diagram Fig. 1, which for clearness sake is not drawn to scale.

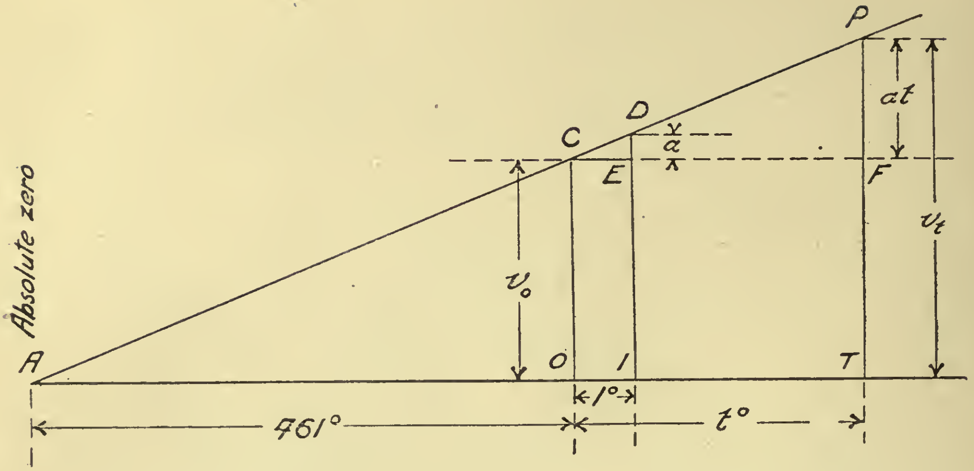


FIG. 1

Let A O T be a graduated scale of temperatures, of which the point O represents 0° F., the point I represents 1° F., and the point T represents t° F.

Let the height of any point on the line A P above A T represent the volume at the different temperatures represented on A T of a mass of air, whose volume is one cubic foot at 0° F. O C represents one cubic foot.

Let I D represent the volume at 1° F. Then E D represents the expansion of one cubic foot of air owing to its being raised in temperature from 0° to 1°. This volume is denoted by "a" and is numerically equal to the "co-efficient of expansion."

The law governing the expansion is such that C P is a straight line. The point A where C P cuts A T represents the absolute zero of temperature.

It is found by experiment that the length of A O is about 461 times the length of O I, so that by geometry

$$\frac{ED}{OC} = \frac{CE}{AO} = \frac{1}{461} = a$$

the co-efficient of expansion. Note carefully that the co-efficient of expansion is calculated as *the fraction of the volume at* 0° by which the air expands when heated through 1°.

F P is the total expansion of this volume of air due to being raised in temperature from 0° to t°. Obviously

$$\frac{FP}{CF} = \frac{ED}{CE}$$

that is

$$\frac{F\,P}{t^\circ} = \frac{a}{1}$$

therefore

$$F\,P = a\,t.$$

From this diagram it is clear that when *any* volume of air at t°, represented by T P, is reduced to 0°, its volume is then represented by O C.

$$\frac{T\,P}{O\,C} = \frac{A\,T}{A\,O} = \frac{461 + t}{461} = 1 + \frac{1}{461}\,t = 1 + a\,t$$

Hence

$$O\,C = \frac{T\,P}{(1 + a\,t)} \quad \text{or} \quad \frac{T\,P \times 461}{(461 + t)}$$

or

$$T\,P = O\,C\,(1 + a\,t).$$

A mass of air whose volume is Vt_1 at t_1° alters in volume to Vt_2 when its temperature becomes t_2° in the proportion

$$\frac{Vt_1}{Vt_2} = \frac{461 + t_1}{461 + t_2} \qquad (12)$$

or dividing above and below by 461

$$\frac{Vt_1}{Vt_2} = \frac{1 + a\,t_1}{1 + a\,t_2} \qquad (12a)$$

Also

$$\frac{\text{the density of air at } t^\circ}{\text{density of air at } 0^\circ} = \frac{\text{weight of 1 cubic foot at } t^\circ}{\text{weight of 1 cubic foot at } 0^\circ}$$

$$= \frac{\text{weight of } (1 \times \frac{O\,C}{T\,P}) \text{ cubic ft. at } 0^\circ}{\text{weight of 1 cubic foot at } 0^\circ}$$

$$= \frac{Do\,\frac{O\,C}{T\,P}}{Do} = \frac{O\,C}{T\,P} = \frac{1}{1 + a\,t}$$

or density of air at t° =

$$\frac{Do}{1 + a\,t} \quad \text{or} \quad \frac{Do \times 461}{461 + t} \qquad (13) \text{ and } (13a)$$

Weight of Air.

At standard pressure one cubic foot of dry air at a temperature 0° F. weighs 0.0863 pounds, and as just explained, it expands 1/461 part *of its volume at* 0° F. for every degree through which its temperature is raised.

The most convenient method of calculating the weight of a given volume of air at a given temperature consists in determining what its volume would be at the standard temperature 0° F., and multiplying by the weight of one cubic foot of air at that temperature, 0.0863 pounds.

The following two examples will show the application of these formulæ :—

Example 1.—Calculate the volume of 255,000 cubic feet of air measured at 40° when raised to 70°.

Let v_{70} be the volume at 70°. Then

$$\frac{v_{70}}{255{,}000} = \frac{461 + 70}{461 + 40}$$

Hence

$$v_{70} = 255{,}000 \times \frac{531}{501} = 270{,}000$$

Example 2.—Calculate the weight of 575,000 cubic feet of air at 85°.

Here the volume of 575,000 cubic feet at 85° if reduced in temperature to 0° F. would become

$$575{,}000 \times \frac{461}{461 + 85} = 485{,}000$$

As each cubic foot of air at 0° weighs 0.0863, the weight of the whole at that temperature must be

$$485{,}000 \times 0.0863 = 41{,}900 \text{ lbs.}$$

For convenience of calculation, Table V. is given, which shows all the properties of air with which the heating engineer is chiefly concerned.

Expansion of Air due to Changes of Pressure.

In addition to depending on the temperature, the volume of any given weight of air depends also on its pressure. The law governing the changes of volume and pressure in a mass of air *when the temperature is maintained constant* is that the volume is inversely proportional to the absolute pressure. Where P represents the pressure in any given units and V represents the volume in any given units,

$$P\,V = \text{Constant, or } P = \frac{\text{Constant}}{V}\text{, or } V = \frac{\text{Constant}}{P} \qquad (14)$$

The value of the constant can be calculated from any pair of known simultaneous values of the pressure and volume.

These equations are only exactly true of what is known as a "perfect gas." Air is not an absolutely "perfect gas," but it is so near to a perfect gas that the heating engineer need not consider the minute variations.

In order to exhibit graphically the variations of volume and pressure, let the volume of any given mass of air be plotted horizontally, and the pressure vertically, as on Fig. 2. Then the equation to the curve representing the expansion at a constant temperature is

$$p\,v = \text{Constant.}$$

Thus, if the volume is v_1, suppose the pressure is p_1. This is shown by the position of point P_1. If the air is compressed until its volume is v_2, and *if the temperature is kept constant* (but only so) the pressure rises to p_2 where

$$v_1\,p_1 = v_2\,p_2.$$

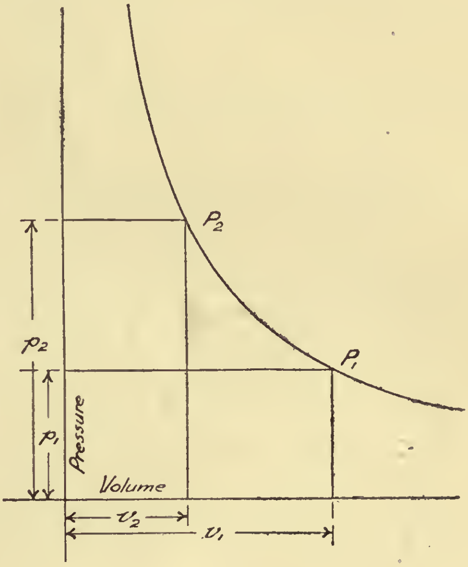


FIG 2

The state of the air is then shown by point P_2.

The curve exhibiting this relation graphically is what is known in mathematics as a "rectangular hyperbola," the shape of which is shown on Fig. 2. If any point P_1 is taken on this curve, the area $O\,X\,P_1\,Y$ is always the same, wherever point P is taken on the curve.

The following are the numerical figures which will enable this result to be applied to any of the ordinary cases with which the heating engineer is concerned.

For one pound weight of dry air, if pressures are calculated in pounds per square foot and volumes in cubic feet, the numerical equation is

$$p\,v = 53.15\,T \qquad (14a)$$

or for m pounds of air

$$p\,v = m \times 53.15\,T$$

where T is the absolute temperature, namely, 461 + t° F.

If pressures are calculated in pounds per square inch, the equation becomes

$$p\,v = 0.369\ T.$$

The corresponding figures for other gases are given in the accompanying table. (Values in foot units are $\frac{53.15}{\text{density rel. to air}}$.)

The value of the co-efficient of expansion is here assumed to be the same for all cases. This is not exactly true, but the variations are very slight. For the purposes of the heating engineer the law may be assumed to be sufficiently accurate for all cases. The actual variation for any case with which the heating engineer will be concerned does not exceed 1/10th of 1%.

The weight of one cubic foot of air at different heights of the barometer may be calculated from the formula

$$\frac{1.3304 \times B}{T}$$

where B is the height of the barometer in inches of mercury column.

Isothermal Compression.

It will be seen that this law only holds *when the temperature of the gas is maintained constant.* It is important to observe that when any gas is forcibly compressed, the mere act of compression involves a considerable expenditure of energy. After the compression the work which has been applied in compressing the gas *resides in the compressed gas in the form of heat.* For every 778 foot-pounds of work that has been applied in the act of compression, one B.T.U. appears in the gas as heat. Thus, compression always raises the temperature of a gas, and until it has been cooled to its original temperature the law $p\,v$ = constant does not hold. The value which $p\,v$ assumes depends on the temperature generated in the gas. For one pound of air it is always $p\,v = 53.15\ T$, and for other quantities in proportion.

Similarly when any compressed gas expands, the work which it does in expansion disappears from the gas. The temperature falls to a corresponding extent.

Adiabatic Compression.

The law connecting pressure and volume in a gas from which no heat is allowed to escape is entirely different from that which holds when the temperature is maintained constant. The former kind of compression is called "adiabatic" and the latter "isothermal" compression. For adiabatic compression, that is, for compression carried out under such conditions that no heat escapes from the gas, the formula is

$$P\ V^{1.408} = \text{Constant.} \qquad (15)$$

The value of the constant for any given conditions can be calculated from the equation above, namely,

$$p\ v = 53.15\ T$$

as in the following examples.

EXAMPLE 1.—It is required to find the relation between the pressure and volume during adiabatic compression of a mass of air, which at 14.7 pounds per square inch has a volume of, say, 13.5 cubic feet. Here $P = 14.7 \times 144 = 2{,}116.8$ pounds per square foot.

Hence, the value of the constant is

$$2{,}116.8 \times (13.5)^{1.408} = 2{,}116.8 \times 39.1 = 83{,}000$$

and the equation is therefore

$$p\ v^{1.408} = 83{,}000.$$

EXAMPLE 2.—Derive a numerical equation for the adiabatic expansion of 25 pounds of air, which at a pressure of 100 pounds per square inch above atmospheric has a temperature of 150° F.

From the equation No. 14 it is clear that the value of $p\ v$ for this mass of air under the given conditions is

$$= m \times 53.15 \times T = 25 \times 53.15 \times (150^\circ + 461^\circ) = 25 \times 53.15 \times 611 = 812{,}019$$

and therefore the volume at 100 lbs. per square inch ($= 114.7 \times 144 = 16{,}500$ lbs. per sq. ft. absolute) and 150° F.

$$= \frac{812{,}019}{114.7 \times 144} = 49.1 \text{ cubic feet}$$

whence the value of the constant is

$$16{,}500 \times 49.1^{1.408} = 16{,}500 \times 240.5 = 3{,}980{,}000.$$

Temperatures of Compression.

The temperature which air will attain when compressed or expanded adiabatically may be found as follows :—

If air at $p_1\ v_1$ is compressed adiabatically $p_2\ v_2$ we have

$$p_1\ v_1^{1.408} = p_2\ v_2^{1.408} \quad \text{or} \quad \frac{p_1}{p_2} = \frac{v_2^{1.408}}{v_1^{1.408}}$$

but

$$\frac{p_1\ v_1}{p_2\ v_2} = \frac{53.15\ T_1}{53.15\ T_2} \quad \text{or} \quad \frac{p_1}{p_2} = \frac{T_1}{T_2} \times \frac{v_2}{v_1}$$

hence substituting

$$\frac{v_2^{\ 1.408}}{v_1^{\ 1.408}} \times \frac{v_1}{v_2} = \frac{T_1}{T_2}$$

whence $$\left(\frac{v_2}{v_1}\right)^{0.408} = \frac{T_1}{T_2} \tag{16}$$

Similarly

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{0.29} \tag{17}$$

EXAMPLE.—Calculate the temperature attained when a mass of air at 50° F. is suddenly compressed to 5 atmospheres.

Here

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{0.29} = 5^{\ 0.29} = 1.6$$

$$T_2 = (461 + 50) \times 1.6 = 816$$

$$t_2 = (816 - 461) = 355^\circ \text{ F.}$$

The relation of the two methods of compression will be best understood by an inspection of the diagram Fig. 3. In this case it is assumed that two equal volumes of air, each V cubic feet, are taken at atmospheric pressure, and one is compressed isothermally and the other adiabatically, and the pressure of each at different volumes plotted on the same diagram.

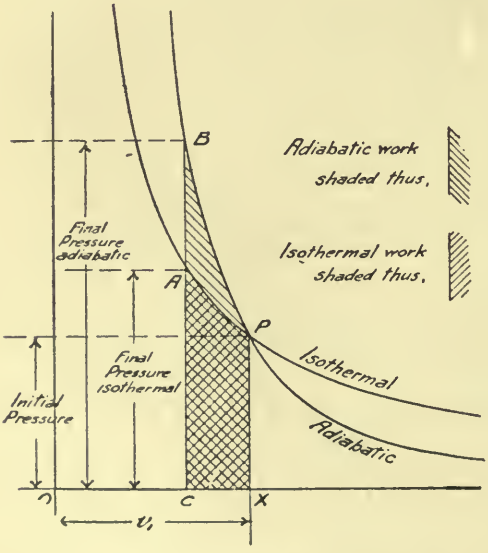

B
Adiabatic work shaded thus,
Final Pressure adiabatic
Isothermal work shaded thus,
A
P
Final Pressure isothermal
Isothermal
Initial Pressure
Adiabatic
O
C
X
v,

FIG 3

In the latter case, as the heat is not allowed to escape but accumulates in the gas, it is evident that after compression the pressure will be higher for the same volume than in the former.

On the other hand, if the air is allowed to expand, the cooling due to the adiabatic expansion will cause the pressure to be less than in the case of the isothermal curve. In other words, the adiabatic curve crosses the isothermal curve at a steeper angle. The work which it is necessary to do in order to effect the compression in these manners is represented by the areas X P A C and X P B C respectively under the curves.

Proof of this may be obtained from any book on thermo dynamics. Here it will suffice to give a formula whereby the amount of work can be determined. These formulæ are as follows :—

Work done in compressing isothermally one pound of air from a pressure P_1 to P_2 is (P in pounds per square foot, V cubic feet, $P\ V = 53.15\ T$).

$$W = P\ V \log_e \frac{P_2}{P_1} \tag{18}$$

Work done in compressing one pound of air from P_1 to P_2 adiabatically.

$$W = 3.45\ P_1\ V \left\{ \left(\frac{P_2}{P_1} \right)^{0.29} - 1) \right\} \tag{19}$$

Work done in compressing one pound of air from V_1 to V_2 isothermally.

$$W = P\ V \log_e \frac{V_2}{V_1} \tag{20}$$

Work done in compressing adiabatically one pound of air from temperature t_1 to t_2.

$$W = 183.45\ (t_2 - t_1). \tag{21}$$

It is important to observe that the mere fact of air being compressed does not imply that the air has any more energy than it has when uncompressed. If the temperature is the same it has not. If compressed air is allowed to do any work, that work is produced entirely at the expense of the heat of the air. If compressed air is allowed to expand *without* doing any work (as when it is allowed to expand into a vacuum), its temperature does not fall at all.

All the above formulæ, strictly speaking, apply when the air is dry, that is to say, when it contains no water vapour. As will presently be shown air always does contain water vapour unless it is specially dried, but the alteration in weight produced by ordinary quantities of water vapour is not great enough to be of considerable importance at temperatures with which the heating engineer has to do.

Specific Heat of Air.

The amount of heat which must be added to one pound weight of air in order to raise its temperature by 1° is

0.1688 B.T.U. = 130.3 foot-pounds

at a constant volume, that is to say, if the air is not allowed to expand during the process. If, however, the air is allowed to expand, the pressure being maintained always the same during the process of heating, the amount of heat required is

0.2377 B.T.U. = 183.4 foot-pounds.

The difference between these two quantities of heat = 53.1 foot-pounds, is due to the fact that if air is allowed to expand while it is being heated, as it must do when heated at constant pressure, it does a certain amount of work in forcing back the surrounding atmosphere, and, according to the principle of the conservation of energy, this work is represented by the disappearance of heat. Thus, the exact quantity of work done during the expansion is the difference between the two quantities of heat, namely,

.0689 B.T.U. = 53.1 foot-pounds per pound of air per degree.

To appreciate the reality of these figures, calculate the actual amount of work done during the expansion of 1 lb. of air while being heated from 62° to 63°.

The tables show that 1 lb. of air at this temperature and at the standard pressure of 14.7 lbs. per square inch (= 2,116.8 lbs. per sq. foot) is 13.141 cubic feet.

In heating from 62° to 63° this volume will expand to

$$13.141 \times \frac{461 + 63}{461 + 62} = 13.141 \times \frac{524}{523}$$

The total expansion will therefore be

$$13.141 \times \frac{524}{523} - 13.141 = 13.141 \times \frac{1}{523}$$

$$= 0.0251 \text{ cubic foot.}$$

The work done in pressing back the atmosphere at a pressure of 2,116.8 lbs. per square foot will be

$$0.0251 \times 2,116.8 = 53.1 \text{ foot-pounds.}$$

This quantity of heat disappears during the process of heating under constant pressure, being converted into work.

It is only the remainder of the heat (0.1688 B.T.U. or 130.3 foot-pounds) which is used in actually raising the temperature of the air, and this quantity is the same whatever the pressure at which the air is maintained.

This is the true specific heat of the substance air. As air is almost always heated at a constant pressure by the heating engineer, the other is the value which must be used in general calculations.

It requires the same amount of heat to raise one pound of air confined in a closed cylindrical vessel at a pressure of say 100 pounds per square inch through one degree as it does to heat the same weight of air confined in a cylinder at 5 pounds per square inch, or any other pressure.

CHAPTER IV.

PROPERTIES OF WATER AND STEAM.

WATER is a colourless, tasteless, odourless substance, which is known in the solid, liquid and gaseous states at temperatures with which the heating engineer has continually to do.

Chemically described it is the product of the combustion or combination of hydrogen and oxygen, in the proportion of 2 volumes of hydrogen to 1 of oxygen, or, by weight, one pound of water is composed of about 1/9 of its weight of hydrogen, and 8/9 of oxygen.

Behaviour of Water under Changes of Temperature.

When a mass of liquid water is reduced in temperature below 32° it becomes solid and turns into ice, giving up during the process of conversion from the liquid to the solid state 142.6 units of heat per pound, which is known as "latent heat." On again raising the temperature above 32° F., the mass of ice again becomes liquid, absorbing in so doing the same quantity of heat as was given out during the opposite process.

Properties of Ice.

Ice cannot exist as ice at a temperature above 32° F.

The temperature at which it melts is slightly altered by an alteration of pressure. Professor James Thompson showed that the melting point is lowered by an increase of pressure, and is slightly raised by a diminution of pressure. The alteration is very small in any case, being something like 0.0133° F. per atmosphere, 14.7 pounds per square inch. According to this, ice would be melted at a temperature of 31° if the pressure were 75 atmospheres, or 1,100 pounds per square inch. This, again, is not a phenomenon with which heating engineers have much to do.

The specific heat of water being 1.0, the specific heat of ice is 0.504. The ratio of the volume of ice to that of the water from which it is formed is 1.086. One cubic foot of ice weighs 57.5 lbs. One pound of ice = 30.1 cubic inches.

Newly fallen snow, which is really crystal of ice in a flaky form, has 12 times the volume of the water formed when it melts. The apparent specific gravity of snow is therefore 1/12.

Specific Heat of Water.

The expression "specific heat" as applied to the substance water when it passes through the temperature 32° is meaningless, because a considerable amount of heat is absorbed at 32° without any change of temperature at all.

A further application of heat raises the temperature of water almost, but not quite, uniformly for equal increments of heat. The specific heat of water increases slightly as the temperature is raised, being 1.00 at 32° F., 1.01 at about 90° F., 1.02 at 130° F., 1.03 at 170° F., 1.04 at 190° F., 1.05 at 212° F., 1.06 at 240° F. (see diagram, Fig. 3*a*).

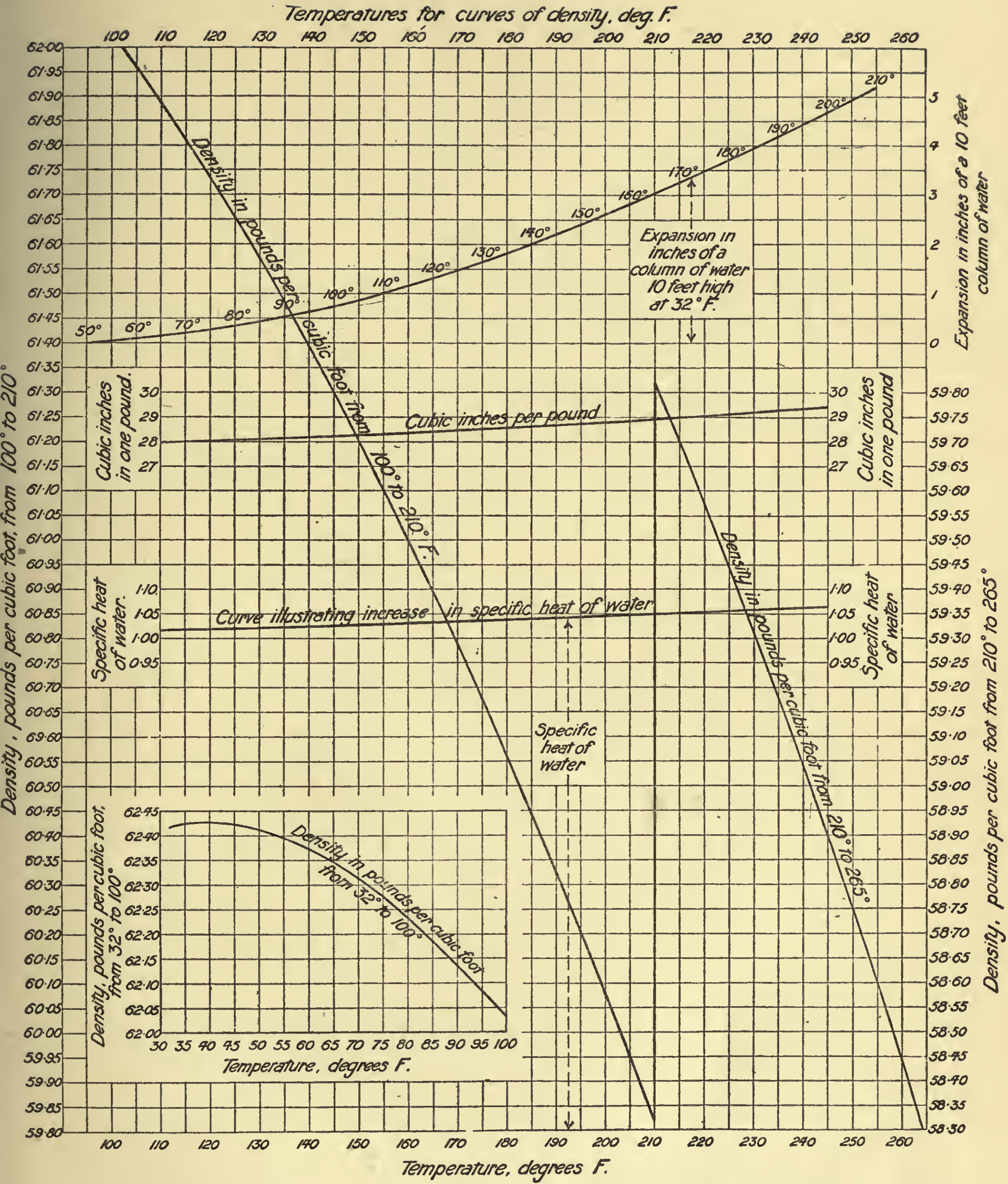
Expansion of Water.

The density of fresh water at different temperatures and its rate of expansion between 32° and 500° F. is given in the accompanying table and diagram, from which it will be seen that about 39° F. is its "point of maximum density."

At temperatures just above freezing point, that is from 32° to about 40°, water *contracts* on being heated, instead of expanding, as do most other bodies. But from 40° upwards, any addition of temperature causes a small expansion of water.

The importance of this phenomenon in nature can hardly be overrated. Water is specifically heavier at 39° F. than at any other temperature. When, therefore, the temperature of the air becomes very low, water at this "point of maximum density" will sink to the bottom of any pool or lake and protect all aquatic life from destruction by frost, unless the frost is sufficiently severe to solidify the whole into a mass of ice. Ice being specifically lighter than water, naturally floats on the surface. If, therefore, any frost is to reach fishes or aquatic animals at the bottom of the water, the heat contained in this mass of water at 39° F. must be conducted through layers of water, and through a solid mass of ice to the colder air above.

Water is an extremely bad conductor when heat is conducted through it in a direction contrary to that in which the convection currents, due to difference of density, tend to carry the heat. In the case of water between 32° F. and 39° F., the higher the temperature the greater is the density.



CURVES ILLUSTRATING EXPANSION, DENSITY, ETC. OF WATER

A.H. Barker, delt.

FIG. 3a.

Convection currents are therefore formed in a direction contrary to that of the flow of heat. Every possible resistance is offered by this beautiful natural mechanism to the escape of heat from water at 39° to colder air above it.

If it were not for this property of water every winter's frost would practically exterminate the whole of aquatic life.

For ordinary purposes the density of water is taken as 62.4 pounds per cubic foot. Thus, when a pressure of 1 inch of water column is spoken of, the water is assumed to have this density unless (as in the case of hot water circulation) there exists some reason for assuming a different standard.

The exact weight of a cubic foot of water at 62° F. (which for many purposes is its standard temperature) is 62.355 pounds, or 998.8 ounces. It is generally taken at 1,000 ounces. The temperature at which the density of water is exactly 62.4 pounds per cubic foot is 52.3° F.

A gallon of water at 62° F. = 0.160 cubic foot = 277.3 cubic inches, weighs 10 pounds. One cubic foot of water = 6.24 gallons.

The pressure of a column of water of height H feet at any temperature is given in pounds per square foot, by multiplying the density of water at that temperature in pounds per cubic foot by H.

The viscosity of water is considerably reduced as the temperature is raised. Water flows more freely through a pipe at a high than at a low temperature. Frictional constants which are accurate for cold water are not so for hot water.

Composition of Waters found in Nature.

The composition of the impurities in the water used by the heating engineer is an important matter, as it affects the interior of the pipes and boilers, etc., with which he has to do.

Water is one of the most efficient solvents known in nature and readily dissolves many solids with which it comes into contact. Therefore, when water is taken from a spring or river it invariably contains considerable quantities of salts in solution.

In ordinary circumstances the heating engineer does not use a large quantity of water, that which he employs being generally unchanged. Thus, in a hot water heating apparatus the same water is constantly used, and only a very small quantity escapes by leakage, evaporation, and in other ways.

The accompanying table shows the composition of scale deposited from water from various districts in England.

The importance of this matter is chiefly in connection with hot water supply apparatus and with steam boilers fed with fresh water.

There are certain kinds of water, chiefly very soft or acid in reaction, which have a very detrimental effect on wrought iron boilers or pipes, so much so that it is necessary either to renew the latter frequently or to supply them in copper or brass, or some less corrosible metal than wrought iron.

Pure soft water, such as condensed water from a steam system, has, in certain circumstances, an extremely bad effect on wrought iron pipes. In many cases pipes have to be renewed within 18 months or two years. The deleterious effect in this case is due to the presence of air in the pipes.

Condensed water pipes as a rule do not run full, so that part of the pipe is occupied by air and vapour and part by water. The pipes tend to corrode badly along the surface where air and water and steam are in contact.

Quite another effect is that produced by very "hard" water, which, when employed in hot water service apparatus, frequently leads to the complete blocking up of pipe by deposit.

Hardness of Water.

When water contains solids in solution it is said to be "hard," otherwise it is "soft."

The hardness in ordinary water may be divided into two parts, which are called (1) "temporary hardness" and (2) "permanent hardness." The "temporary hardness" consists largely of calcium or lime salts, which are more soluble in cold water than hot, which are therefore deposited or precipitated when the water is heated to a certain degree. This deposit does not always take place in the boiler, where one would expect, although a large part of it does. The circulation of the water into the pipe carries the precipitated matter with it, and it is ultimately deposited in the flow pipe, which often becomes completely blocked. In many cases, however, the deposit takes place on the interior surface, leaving a continuous passage of perhaps a ¼ or ½ inch diameter through a long pipe, say 1½" internal diameter. This interior core does not always become blocked, although the efficiency of the apparatus is, of course, much reduced. Permanent hardness of water does not cause deposit in boilers or pipes, unless the water carrying the solids in solution is actually evaporated away as in steam boilers.

In order completely to get rid of the tendency to deposit solids in solution, it is necessary to soften the water before use, and this is a fairly expensive and inconvenient process. This process consists of mixing with the water a calculated proportion of some chemical which causes the dissolved solids to precipitate. The insoluble precipitate is then filtered off before the water is used. The solids left in solution are such as will not cause a hard deposit. The deposit of temporary hardness may often be reduced or altogether avoided by omitting to heat the water to a high temperature and by keeping the heating coil or surface at a relatively low temperature. Frequently a large part of the deposit contained in water is not precipitated before a temperature of 160° to 180° is reached. If arrangements are made whereby the water is never heated beyond, say, 130° the deposit does not take place to anything like the same degree. This may frequently be done by means of a secondary system of heating, of which more will be explained in a later chapter.

The degree of hardness of water is specified in grains per gallon. Water is said to have one "degree of hardness" when it contains one grain of bi-carbonate or sulphate of lime per gallon. Each degree of hardness destroys one ounce of soap in 40 gallons of water, that is to say, 40 grains of bi-carbonate or sulphate of lime destroys one ounce of soap. 6° of hardness, or under, is reckoned as "soft," above this quantity "hard water."

The effect of the solution of certain ingredients in water on lead pipes, with which they are brought into contact, is to impregnate the water with lead. This is very dangerous where such water is used for drinking purposes.

Character of Deposit from different Qualities of Water.

In order to show the chemical constituents of the deposit from various classes of water the accompanying table is given in the Appendix.

Vapour of Water.

The vapour of water plays an extremely important part in the composition of the atmosphere, and a part hitherto not generally recognised by the ventilating engineer. It will, therefore, be necessary to deal with the matter in some detail.

Water, as we know it in practice, is always subjected to the pressure of the atmosphere, whose magnitude, as we have seen, is 14.7 lb. per square inch, or 2,116.8 lbs. per square foot. Air at about this pressure is always present close to the surface of cold water. Its presence masks and renders difficult of comprehension the operation of the natural laws governing the generation of vapour.

For the purpose of clearing our ideas let us first examine the conditions of the heating of water in a perfectly air-tight vessel, from which all air has been removed.

Such a vessel protects the water from the atmospheric pressure and we can therefore consider the generation of vapour quite apart from the air pressure.

At whatever temperature the water exists in the vacuum chamber its surface continually gives off vapour or water in a gaseous form. In other words, particles of the water detach themselves from the surface and fly about in the otherwise empty space at a great velocity. There are so many of these particles, and they are dashing about in this empty space at such a great speed, that they continually collide in immense numbers with one another, with the walls of the chamber, and with the free surface of the water. They thus jointly produce the effect of a continual pressure on the walls. This is, of course, the physical explanation of the pressure of any gas. We can only appreciate the effect of these impacts by the pressure effect they jointly produce.

In the case of a vacuous space, partly filled with a liquid like water, the greater the temperature the greater is the number of particles which detach themselves from the surface in a unit of time. When the temperature is kept constant at any particular value the particles flying about in the space eventually become so numerous that just as many collide with the surface of the water and adhere to it as those which detach themselves.

There is a continual exchange going on, new particles are continually leaving the surface and an equal number are splashing into it.

When things are in this condition the space is said to be "saturated" with vapour. The vapour is in a sort of equilibrium with its own liquid.

A constant condition is thus set up in which the number of particles flying about in the space becomes constant for that temperature.

When this state of things is established the pressure which is maintained on the walls of the chamber is constant just so long as the temperature remains constant. When the temperature increases the number of particles in circulation and the resulting pressures both increase at the same time. There is a definite maximum pressure corresponding to every value of the temperature, which can be easily measured by a delicate manometer.

The value of this pressure is shown for all temperatures in the diagram No. 3*b*. At a temperature of 212° F. this pressure becomes equal to the normal atmospheric pressure, 14.7 lbs. per square inch.

Now if the temperature is maintained at *any* given value above or below 212° the vapour pressure being at its corresponding value, and if the pressure is suddenly relieved by any means, such a large number of water particles rush out of the body of the water through the surface, not being hindered or balanced by a corresponding number of returning particles, that the surface of the water is thrown into more or less violent agitation. The process of vaporisation, hitherto proceeding quietly, now becomes explosive—the water "boils."

The violence of the process increases in proportion to the magnitude of the discrepancy between the actual reduced pressure and the maximum vapour pressure corresponding to the temperature.

The vapour pressure in the interior of the liquid is great enough to support not only the pressure on the surface of the liquid, but a column of fluid as well; in other words, bubbles of steam are formed. Vaporisation proceeds at the interior surface of the bubble. The bubble rises owing to its lightness and explodes at the surface of the water.

Vapour cannot exist at a temperature lower than that corresponding to its pressure. If it is subjected to such a temperature, part of the vapour immediately condenses until the pressure exerted by the remainder corresponds to the maximum possible vapour pressure at that temperature.

The rate at which the vapour is formed in an unsaturated space depends, as has been seen, on the difference between the pressure corresponding to the temperature of the water and the actual pressure on the surface of the water.

The difference between what is called "vapour" and steam is therefore seen to be one of degree only. Steam is only called vapour when it is mixed with air, or when it is below its saturation point, but the distinction between vapour and steam is not at all definite. Steam is rarely called vapour above the pressure of the atmosphere.

When steam is at the natural temperature of the water from which it is produced, it is also said to be "saturated." If its volume is increased when in this condition, without any heat being supplied to it, its pressure is reduced. The energy expended in forcing back the atmospheric or other pressure is derived from the conversion into water of some of the steam ; in other words, some of the steam will condense.

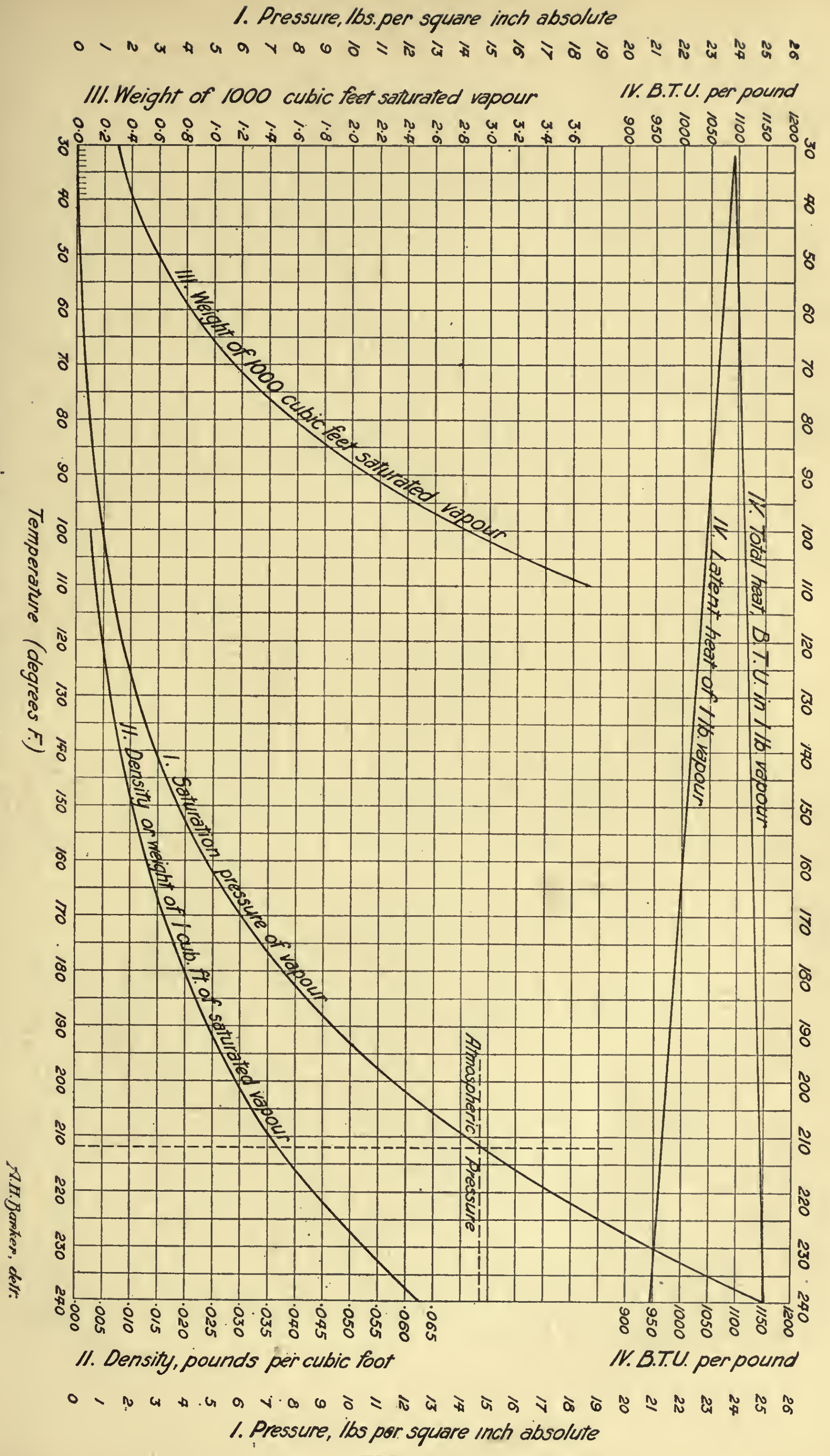
Steam in contact with water always rapidly becomes saturated. If heat be added to steam, out of contact with water, the steam rises in temperature, and tends to assume more and more the physical properties of a permanent gas. It is then said to be "super-heated." If saturated steam is compressed it becomes super-heated, the energy expended during the process of compression is converted into heat, which is absorbed by the steam, thus raising its temperature.

The diagram shows, in graphical form, all the properties of steam with which the heating engineer need make himself familiar.

At a very high temperature, above say 600° F., steam is said to disintegrate into its constituent gases, oxygen and hydrogen. This is a phenomenon with which the heating engineer, as such, has little to do.

Latent Heat.

The process of conversion of water from the liquid to the gaseous state is attended by the absorption of a great deal of heat.



PROPERTIES OF VAPOUR OR STEAM.

FIG. 3b.

Heat has been defined as the amount of molecular energy in a body. When the molecules of water are flying about in the form of gas they might be expected to contain more of this miscellaneous kinetic energy than when they are relatively at rest in the form of liquid. This is found to be the case in practice.

Thus, every pound of water which is converted into vapour at atmospheric pressure absorbs about 966 B.T.U. The amount of heat absorbed increases as the pressure is reduced.

After vapour or steam is formed it is again condensed by being brought into contact with some surface colder than itself at a constant pressure, giving up to the cold surface the heat which it has acquired during the process of vaporisation. This is the property which makes steam such a valuable medium for carrying heat.

Total Heat of Steam.

What is called the "total heat of steam" is not in reality its total heat at all. It is only that portion of the total heat reckoned above an arbitrary datum level. Strictly speaking, the "total heat of steam" would be the total molecular energy in the steam above absolute zero. As the total molecular energy below, say 32°, is not accurately known, and as this portion of the total heat of steam is never used in practice, this temperature, 32°, is generally reckoned as an arbitrary datum level.

The heat which is communicated to 1 lb. of water in order to secure its conversion into vapour is composed of several parts when reckoned above ice at 32° F.

(1) The latent heat required to melt the ice.

(2) What is called the "sensible heat," which is required to raise the temperature of the water from 32° F. to 212° F., or any other temperature corresponding to the pressure of steam required. The effect of this portion of the heat is to increase the velocity of the molecules and expand the liquid. During this process of heating a certain minute amount of the heat is absorbed in doing work by pushing back the atmospheric pressure.

(3) The latent heat which is employed in overcoming the attraction of the molecules for one another, and in communicating to them the kinetic energy which they have in virtue of the velocity with which they fly about in the space occupied by the steam.

Part of this latent heat is absorbed in doing work in forcing back the atmospheric pressure, or any other pressure, to which the steam is subjected, such as the pressure on the piston of a steam engine.

In the case of steam generated at atmospheric pressure the number of foot-pounds of mechanical work done in evaporating one pound of water may be calculated as follows :—

Volume of one pound of steam at atmospheric pressure
26.4 cubic feet.

Atmospheric pressure, 14.7 pounds per square inch = 2,116.8 pounds per square foot.

Work done = 2,116.8 × 26.4 = 55,700 foot-pounds, or 72 B.T.U.

The following table shows the totals for 1 lb. of ice melted, heated to 212° and evaporated at that temperature :—

Operation.	Heat absorbed.	Equivalent mechanical work, 1 B.T.U. = 778 ft. lbs.
Melting 1 lb. ice	142.6	111,000
Heating water 32° to 212°	181	140,500
Generating Steam—		
(*a*) Mechanical work against atmospheric pressure ...	72	55,700
(*b*) Overcoming molecular attraction	893	692,000
Total water to steam	1,146	888,200

Super-heated Steam.

When saturated steam is heated to a temperature higher than that at which it is formed, it is said to be "super-heated."

When it is heated in this manner it expands rapidly for the first few degrees above its saturation point, afterwards it expands in just the same way as air, that is as if it were almost a "perfect gas."

When vapour below a temperature of about 150° is heated it expands almost as a perfect gas. Above a temperature of about 175° the expansion for the first 10° is about three times that of air.

The specific heat of saturated steam is 0.305, gradually increasing to 0.475 as the distance from the saturation point increases. In other words, the specific heat of saturated steam is about 1.28 times as great as that of air, which is 0.238.

Specific heat of super-heated steam or vapour is 1.7 times that of air.

The diagram, Fig. 3*b*, on page 47, shows the variation of the total heat, the latent heat, pressure, and weight of steam and vapour for all temperatures from 32° to 240°.

Mixture of Vapour and Air.

When air and water vapour exist together in a closed vessel the joint total pressure which will be exerted on the walls of the vessel is the sum of the pressures of the air and of the water vapour. Each is absolutely independent of the other. The magnitude of the pressure exerted by the vapour alone does not depend at all on the presence or absence of air mixed with the vapour.

Thus, if it is known that a certain weight of dry air is contained in a certain vessel, and if there is also a known vapour pressure in the vessel, it is easy to calculate what the total pressure is on the walls of the vessel, by adding together the calculated pressure of the air and the given pressure of the vapour.

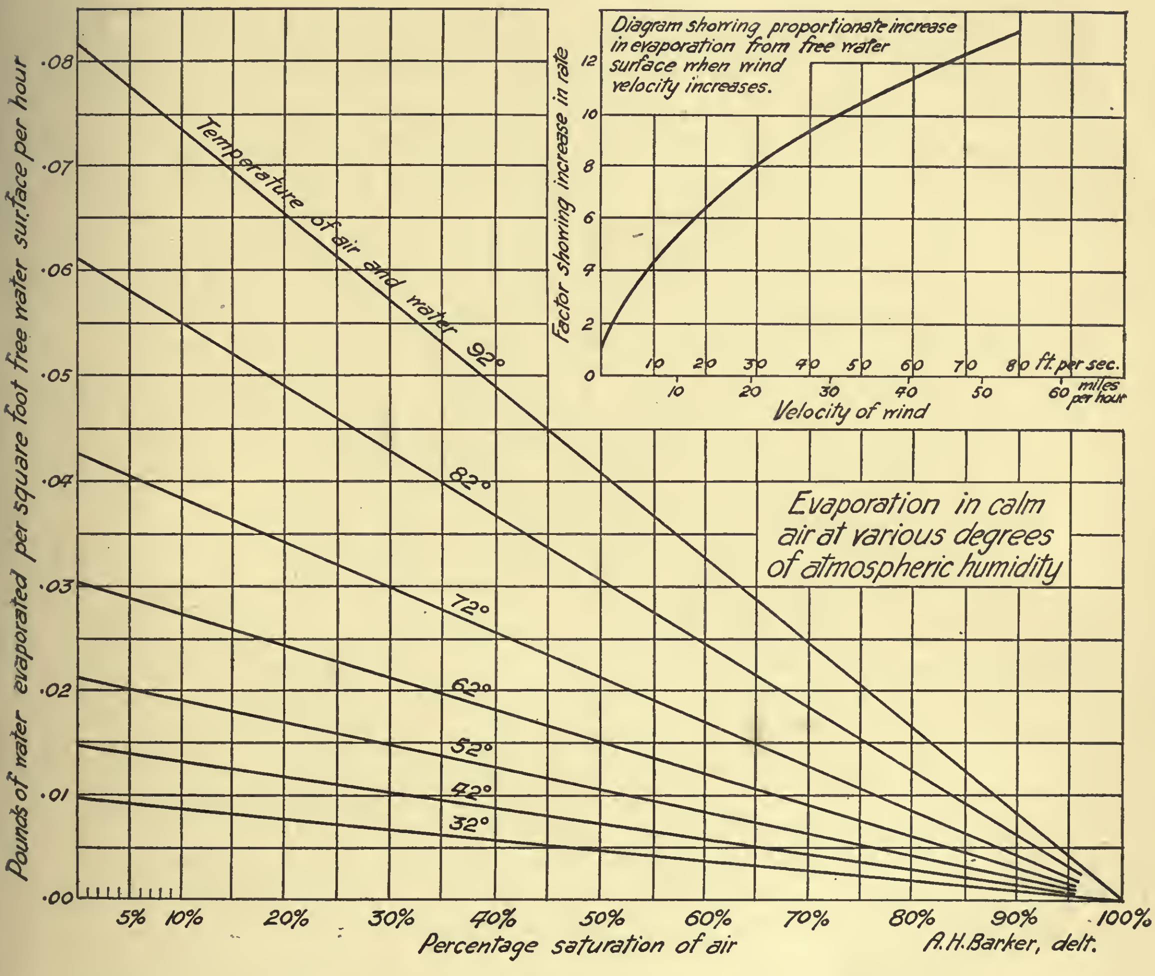
The mixture of air and vapour is, of course, of the most intimate kind. The vapour does not exist in the air as a number of minute drops, but as a gas thoroughly incorporated with the air, just as any two gases may be mixed together in such a way that the composition throughout the whole mass is absolutely uniform. The vapour may be imagined as being dissolved in the air in the same way as salt is dissolved in the sea.

When any given space contains vapour at a pressure corresponding to the maximum for that particular temperature the space is said to be "saturated" with vapour, whether or not there is air in the space as well. It is saturated in this sense, that that particular *space* cannot hold any more vapour at that particular temperature and pressure. If the cubic space in which the saturated mixture is confined is reduced, the temperature being maintained constant, some of the vapour will condense into water, and the amount of vapour existing in the space per cubic inch will be exactly the same as before. Also *if the temperature* in the saturated space is slightly reduced some of the vapour will at once condense into water, and the pressure exerted by the remainder of the vapour will be reduced in a corresponding degree. If the temperature is increased some of the water will pass into vapour (assuming that the space contains some unevaporated water as well as vapour), and the vapour pressure will increase in reality.

It should be carefully noted that in speaking of a quantity of *air* as being saturated with vapour we really mean that *the space occupied by the air* is so "saturated," the presence of the air itself is a mere accompaniment and not an essential to the saturation of the space with vapour.

We can thus calculate the weight of vapour in any saturated space independently of the amount of air there may be in it, but since we are always in practice dealing with the vapour as it exists *in the air* it is a convenient method of expressing ourselves, though it is liable to lead to misconception to speak of the air itself as being saturated with vapour.

The table then gives the amount of vapour by weight, which is sufficient to saturate one cubic unit of space at all temperatures. Observe, again, that it would not be correct to speak of the amount of vapour which would be required to saturate *one pound*, or any given mass of air, but



DIAGRAMS SHOWING RATES OF EVAPORATION FROM SURFACE OF WATER EXPOSED TO AIR AT DIFFERENT TEMPERATURES, HUMIDITIES AND VELOCITIES (Box)

FIG. 3c.

one cubic foot or some other *volume* only, because the amount of vapour which one pound of air at any given temperature will contain depends on the volume only, and only on its pressure in so far as that function is related to its volume.

Air which is not completely saturated with vapour will take up vapour from water which is exposed to its action, in virtue of the fact that the space in which the air is contained is not already fully saturated.

When water is being heated with its surface exposed to the air, since the generation of vapour continually takes place, part at least of the heat applied to the water is absorbed in generating vapour.

The rate at which vapour is produced from a free surface of water depends on the amount of vapour already in the air, and on the rate at which the air above the water is renewed. If it is not renewed at all, *i.e.*, if the water is heated in a confined space, the production of vapour ceases as soon as it attains the maximum pressure corresponding to the temperature. If the air is renewed rapidly (as by a wind) the production of vapour is very rapid, and the heat so absorbed is sufficient to reduce the temperature of the water considerably if its depth is small.

Thus, when a free surface of water is exposed to the open air evaporation continually proceeds, unless the air is already saturated.

Since water vapour (H_2O) is lighter than air $\left(\frac{O_2 + 4N_2}{5}\right)$ air charged with water vapour is lighter than dry air. The more moist the air is the lighter it becomes. The air close to the surface of the water naturally becomes more moist and rises through the drier air above, in just the same way as heated air from a radiator will rise.

Any free surface of water therefore maintains a continual circulation of air which carries some of the water with it. In this way the evaporation is maintained.

At any given temperature and in still air the rate of evaporation is proportional to the difference between the maximum vapour pressure at that temperature and the actual vapour pressure in the surrounding air. The rate of evaporation per square foot in still air at one particular temperature and humidity and with water at varying temperatures is shown on the accompanying diagram, Fig. 3*c*. The large diagram, Fig. 3*d*, shows the figures for still air at various temperatures and humidities, the water being at the same temperature as the air.

The small diagram shows the rate at which the evaporation increases when the air velocity over the surface increases.

It will be seen that in a high wind the rate of evaporation is 12 times as great as in calm air.

The total loss of heat from such surfaces consists of the latent heat

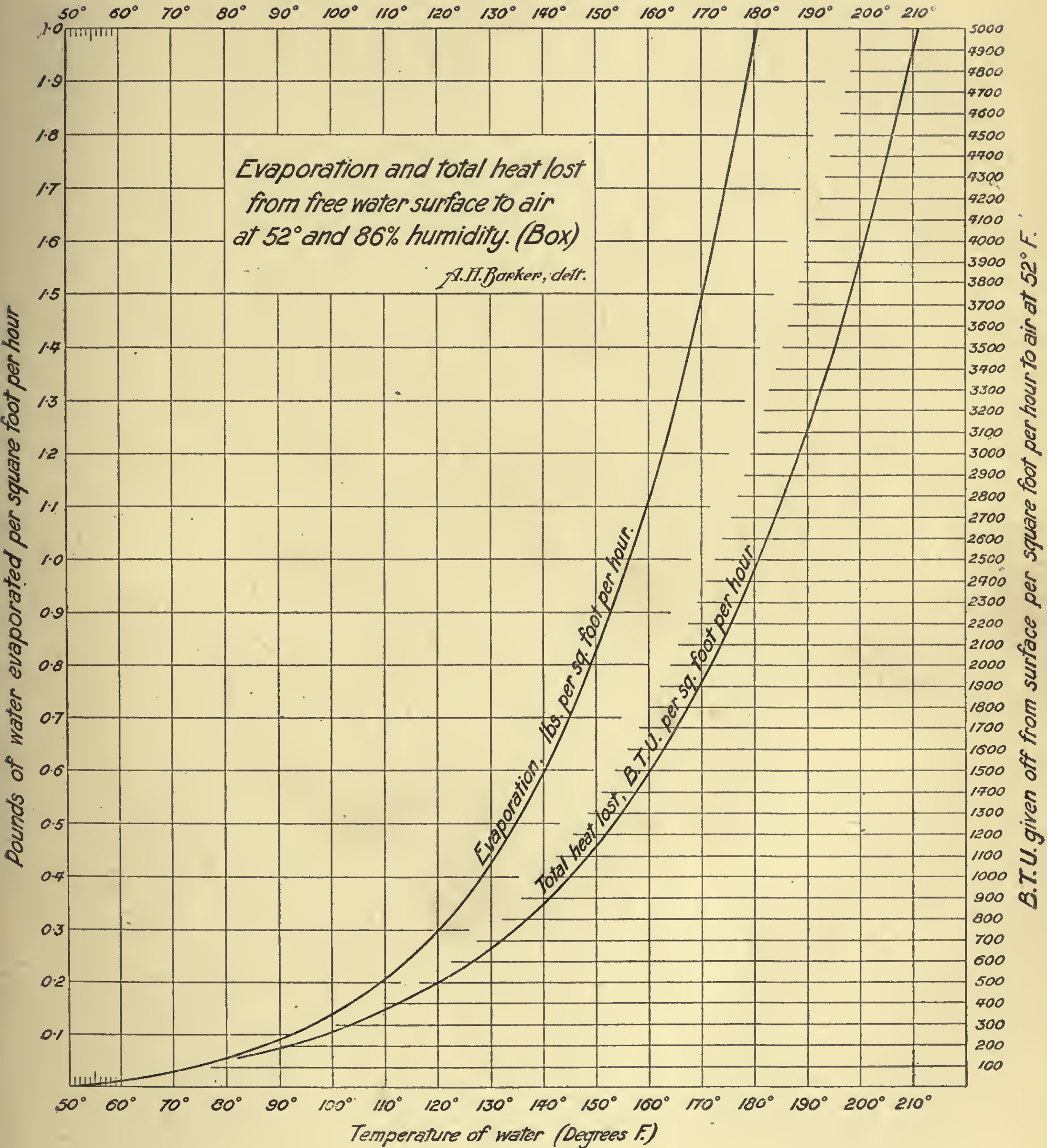

Evaporation and total heat lost
from free water surface to air
at 52° and 86% humidity. (Box)
A. H. Barker, delt.
Evaporation, lbs. per sq. foot per hour.
Total heat lost, B.T.U. per sq. foot per hour
Pounds of water evaporated per square foot per hour
B.T.U. given off from surface per square foot per hour to air at 52° F.
Temperature of water (Degrees F.)
50° 60° 70° 80° 90° 100° 110° 120° 130° 140° 150° 160° 170° 180° 190° 200° 210°
2·0 1·9 1·8 1·7 1·6 1·5 1·4 1·3 1·2 1·1 1·0 0·9 0·8 0·7 0·6 0·5 0·4 0·3 0·2 0·1
5000 4900 4800 4700 4600 4500 4400 4300 4200 4100 4000 3900 3800 3700 3600 3500 3400 3300 3200 3100 3000 2900 2800 2700 2600 2500 2400 2300 2200 2100 2000 1900 1800 1700 1600 1500 1400 1300 1200 1100 1000 900 800 700 600 500 400 300 200 100

FIG. 3d.

of vaporisation + the heat lost from the surface by radiation and contact of air. A curve showing these values is also appended, Fig. 3*c*.

The rate at which the vapour is absorbed also depends on the amount of air present in the space. The greater the air pressure the slower the space takes up the vapour.

Close to the surface of the water the vapour always exerts its pressure, whether it is given off in a confined space or not. If generated in a confined space this pressure is exerted on the interior surface of the containing vessel, and if in the open air the pressure operates against the fluid pressure of the atmosphere.

If the surface of the water is open to the atmosphere, the vapour pressure in the layer of air close to the surface of the water is probably the same as in an enclosed space. If the water is contained in a tank covered with a loose cover, the space above the water becomes saturated as the air cannot get away freely enough to remove the accumulations of vapour.

There is also this difference between the case of water heated in an air-tight space and in the open air. In an air-tight space the confined air cannot get away at all. As the temperature increases the *total* pressure in the vessel rises, owing to the warming of the air as well as to the increase of vapour pressure, as is explained in the chapter on Properties of Air.

In the open air the *total* pressure of the air and water vapour together must of necessity remain constant. In the region near the surface of the water the *total* pressure is, of course, the same as in the surrounding space, but the higher the temperature of the water the greater is the proportion which the pressure of the vapour bears to the pressure of the air in the total, and therefore the greater is the proportional amount of vapour present and the less the amount of air. At greater distances from the free water surface the more does the vapour pressure approximate to the general vapour pressure in the atmosphere.

As the temperature of the water rises a greater and greater proportion of the total pressure near the surface is due to the vapour, until when the temperature of the water becomes such that its vapour pressure is equal to the total pressure of the atmosphere, all the air is driven back from the surface and the whole of the space near the water is occupied by vapour or, as it is then called, "steam."

If the application of heat to the water continues the water *endeavours* to rise in temperature as before, but the moment it actually rises by even a small fraction vapour or steam is given off with such explosive rapidity that all the surplus heat is absorbed in the production of vapour, and the temperature of the water remains constant. The water is then said to "boil."

It will be understood from the preceding explanation that the process of "boiling" water is only associated with the temperature 212° F. when the pressure is 14.7 lbs. per square inch—the normal atmospheric pressure. The generation of vapour goes on at a lower or at a higher pressure in precisely the same way as at the standard pressure. At a lower or higher pressure explosive vaporisation or "boiling" commences, when the temperature is such that the vapour pressure is equal to the given pressure. Below that point the vaporisation takes place quietly from the surface.

Whatever the temperature may be water will always boil if the pressure on its surface is reduced as far as the maximum pressure of water vapour at that temperature.

Thus, on the top of a high mountain water will boil at a much lower temperature than at the sea level.

When water is raised to the boiling point at any particular pressure it continues to absorb heat as it evolves steam, without raising the temperature. The temperature of the steam given off is the same as the temperature of the water from which it is produced.

The heat is therefore called "latent" and exists in the steam. If the pressure is maintained constant, even a fractional reduction of temperature in the steam will restore the whole of the latent heat to any body at a lower temperature with which it comes in contact, the presence of which causes the reduction in temperature.

Measure of Saturation.

In order to obtain a measure of the degree of wetness of the air at any time, it is convenient to express it as a fraction of its degree of wetness when completely saturated.

The air is said to have a "relative humidity" of, say, 60%, when the space occupied by the air holds 60% as much vapour as would be required thoroughly to saturate it. By the aid of the table given this can be reduced to pounds, or the weight may be expressed in any other units.

Dew Point.

If any volume of an unsaturated mixture of air and water vapour be isolated from the rest of the outside air, and the temperature reduced, it is clear that the total weight of water vapour (or water) in the air-tight space must remain constant.

As the temperature is reduced the margin of power which the space has of maintaining the vapour as vapour, is reduced, until such a point is reached that the pressure of the vapour exactly corresponds to the maximum pressure which water vapour can exert at the reduced temperature. At this point the space is "saturated" with vapour, and any further reduction of temperature causes actual condensation of some of the vapour. This point of temperature is called the "dew point." It can be ascertained experimentally and forms a very accurate means of determining the actual quantity of vapour in the air at any given time. If, for instance, when the temperature is 60° F., the dew point is 41° F., we can tell from the tables that at 41° F. the quantity of water vapour which will saturate a cubic foot of space is .00042 pounds. This, therefore, represents the exact amount of vapour actually existing in such a volume of the air at 60°, as when it is reduced to 41° would occupy one cubic foot of space.

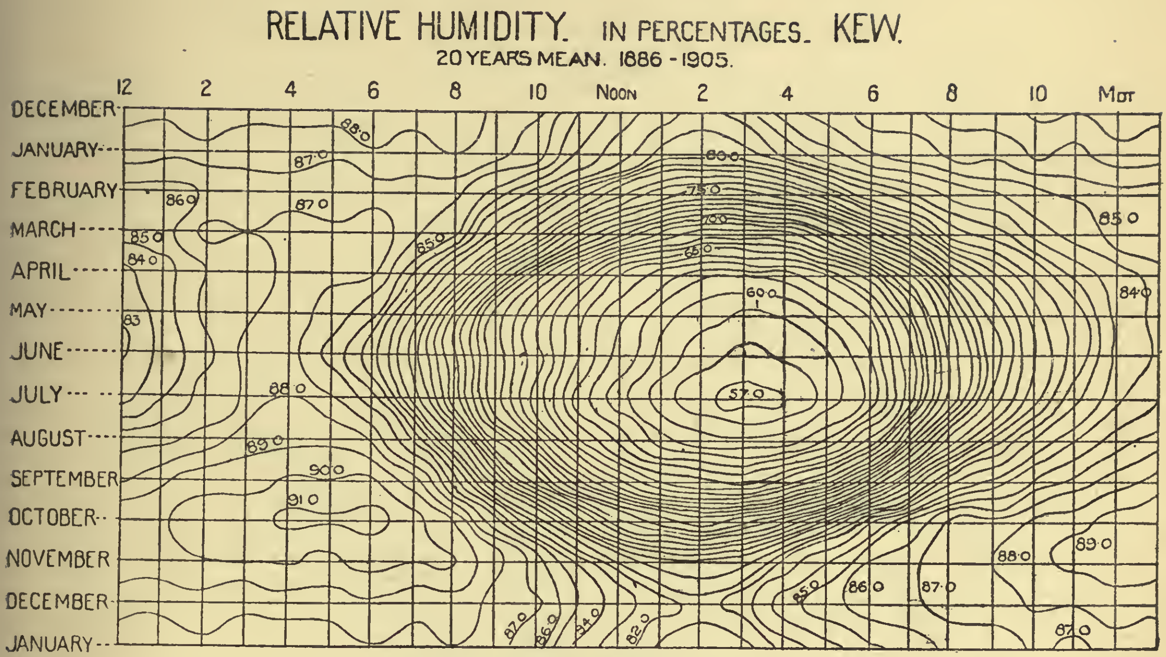
Wet and Dry Bulb Thermometers.

The carrying out of this test, however, requires the possession of considerable skill in chemical manipulation. It is very desirable to obtain a simpler method than this of determining the amount of vapour in air. Such a method is found in the "wet and dry bulb thermometer." This instrument consists merely of two identical thermometers fixed side by side, the bulb of one of which is covered with a thin muslin or lamp wick. The lower end of the fabric dips into a vessel of water. The fabric in contact with the bulb is thereby kept continually moist by capillary attraction.

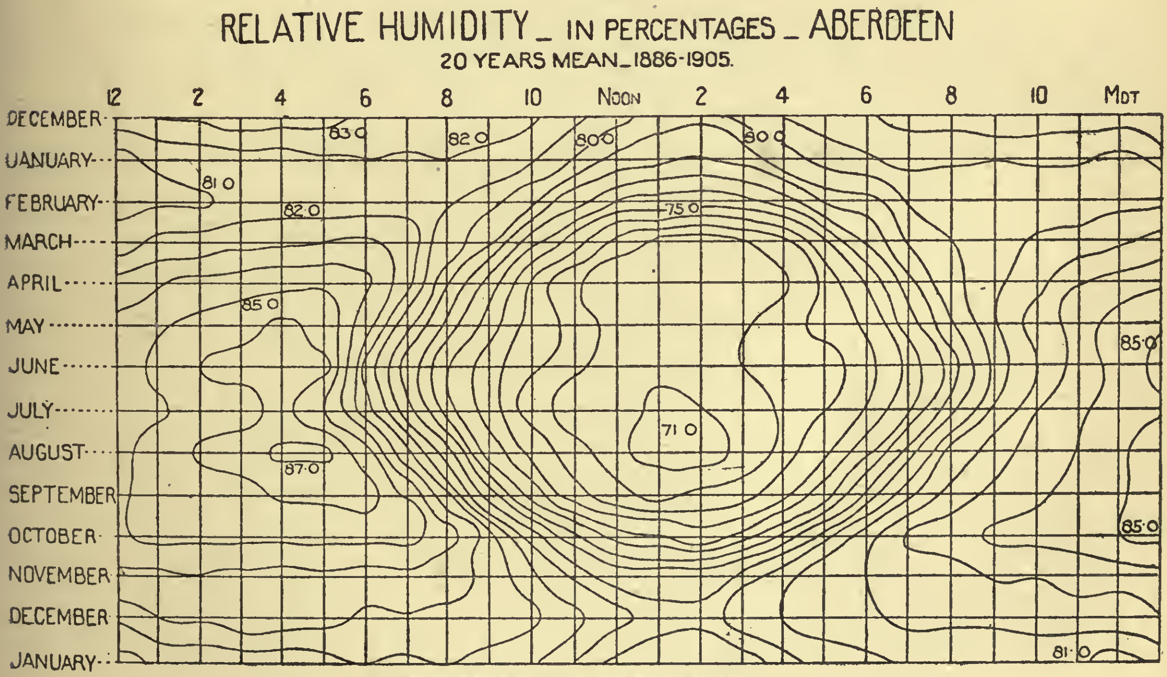
The rate at which the moisture on the fabric evaporates depends on the amount of moisture already in the air. If the air is very wet or nearly saturated with moisture the evaporation will be slow. If, on the other hand, the air is very dry the evaporation will be rapid.

Evaporation of any kind always absorbs heat. The more rapid the evaporation the more heat is absorbed, that is to say, the more rapid the evaporation the greater the degree of cold which is communicated to any body in close contact with the evaporating fluid. In this case, therefore, the dryer the air around the thermometer the more water is evaporated from the muslin, and therefore the colder the bulb of the thermometer is made.

It is found by careful experiment that the degree of cold thus produced is quite constant for a given temperature and dryness of the air. In other words, the wet bulb of a thermometer will always be reduced in temperature by the same number of degrees in all cases where the temperature and the dryness of the air are the same. If, therefore, a careful series of observations are made of the different readings of the two thermometers, under different conditions of temperature and dryness, a series of empirical values can be found and arranged in the form of a table, such that the condition of the air as regards moisture is at once given when the readings of the two thermometers are given. Such a set of observations has been made by Glaisher, and embodied in tables known as "Glaisher's Tables,"


RELATIVE HUMIDITY. IN PERCENTAGES. KEW.
20 YEARS MEAN. 1886 - 1905.

Isopleths showing diurnal and seasonal variation.


RELATIVE HUMIDITY _ IN PERCENTAGES _ ABERDEEN
20 YEARS MEAN _ 1886-1905.

Isopleths showing diurnal and seasonal variation.

and these are given in the Appendix, Table VII., and in the diagram Fig. 3*e*.

The tables are arranged in such a way that the product of the given factor into the difference between the readings of the wet and dry bulb thermometer, deducted from the actual temperature, gives the dew point; in other words, where t' is the reading of the dry bulb thermometer, and t'' that of the wet bulb, and C the co-efficient, then

$t' - C\,(t' - t'')$ is the dew point.

The diagram, Fig. 3*f*, gives the amount of water in the air under all conditions. This diagram is reproduced from Professor Carpenter's work on "Heating and Ventilation," by his kind permission.

In this diagram the ordinate of the curves shows the amount of water in pounds or grains contained in one cubic foot of space, when the temperature is represented by the abscissa, for any degree of saturation. It also shows what the depression of the wet bulb thermometer will be, corresponding to any given degree of saturation.

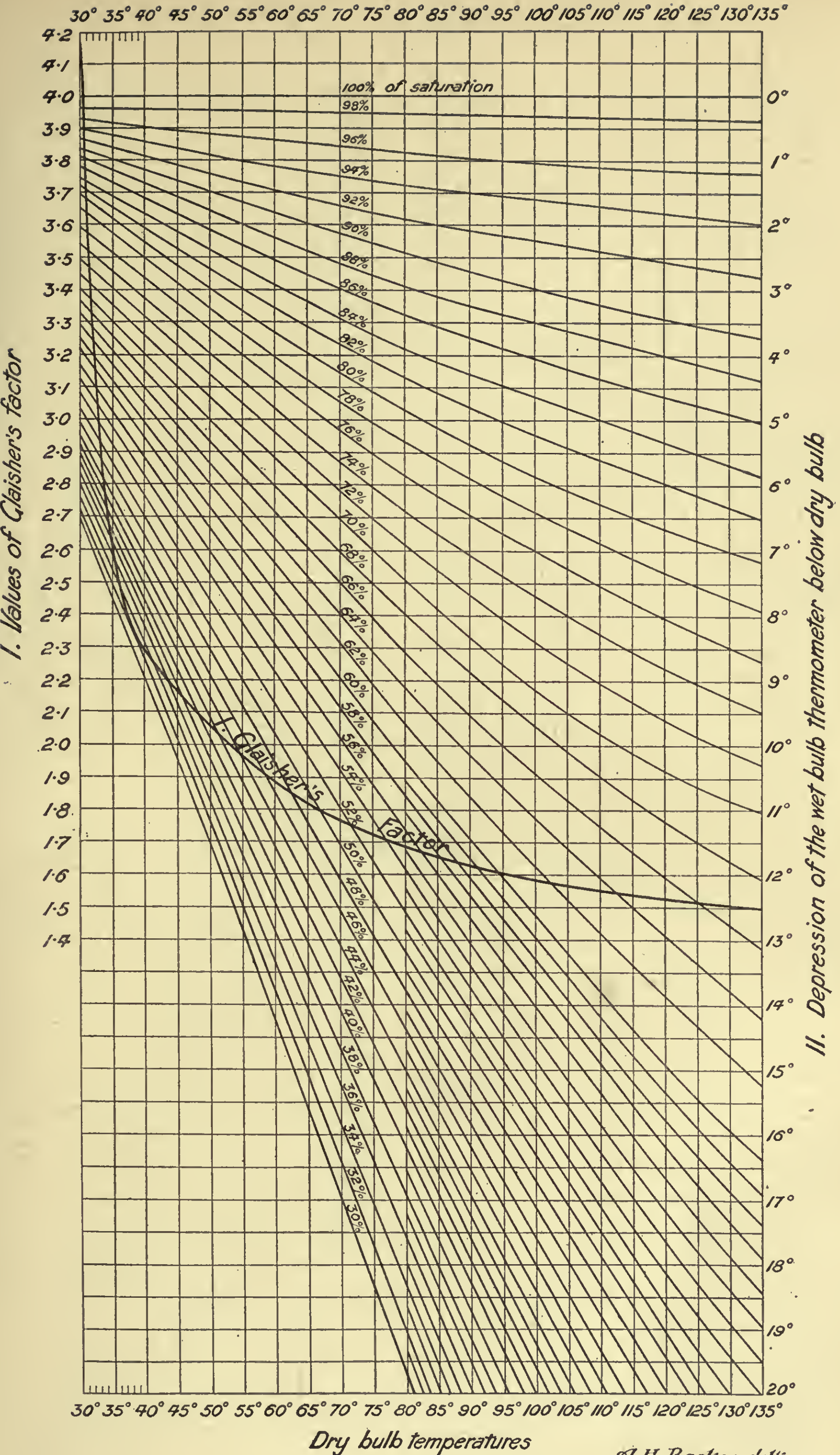
Assume, for instance, that the temperature of the air is 64°, and its hygrometric condition is 50%. To find the quantity of vapour in these circumstances, follow the vertical line representing 64° upwards until it strikes the strong curve representing 50% of saturation, then follow a horizontal line passing through the point of intersection. This shows that the air then contains .00047 pounds of water vapour per cubic foot. It shows also that the temperature of the wet bulb thermometer will be about 53½°, because that point is situated midway between the line sloping down to the right, representing 53° and 54°.

From this beautiful diagram practically any problem relating to the hygrometric condition of the air can be worked out.

Example.—An enclosed mass of air at 70° and having a hygrometric condition of 70% is heated up to 120° F. without addition of moisture. What will be the percentage of saturation then, and what will be the wet bulb temperature? To solve this problem, follow the vertical line representing 70° until it cuts the curve representing 70%. Follow the horizontal line passing through that point and representing .0008 of a pound per cubic foot until it reaches the vertical line representing 120°. This point lies almost on the curve representing 16%, which represents what the degree of saturation would be if the volume of the air did not increase. This figure must therefore be corrected as follows if the air is allowed to expand while being heated.

One cubic foot of air at 70° when heated to 120° at a constant pressure increases in volume to

$$1 \times \frac{461^\circ + 120^\circ}{461^\circ + 70^\circ} = \frac{581}{531} = 1.09 \text{ ft.}^3$$



A. H. Barker, delt.

Curves shewing, (I) Values of Glaisher's factor, and (II) the relative humidity corresponding to various values of the depression of the wet bulb thermometer.

FIG. 3e.

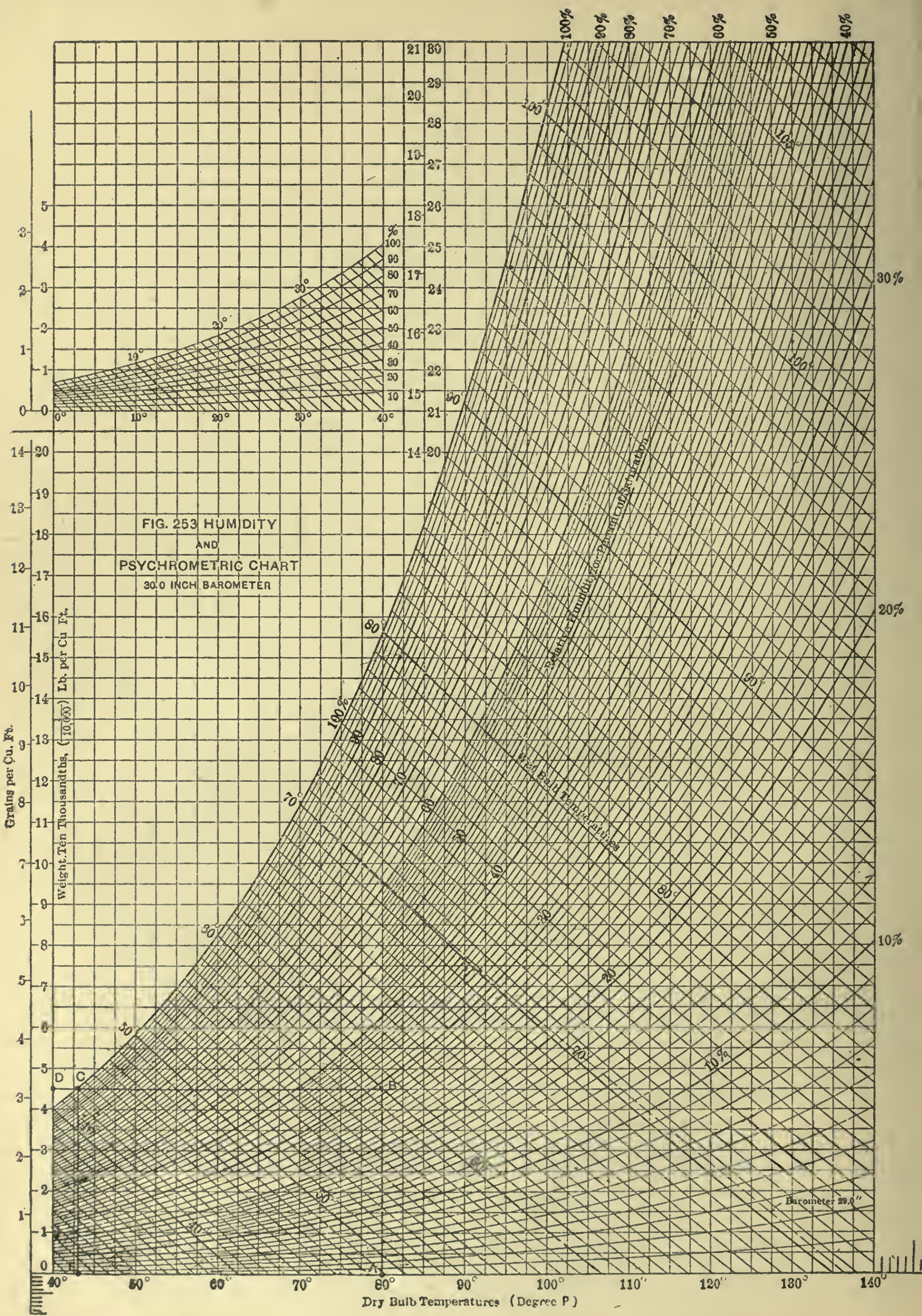


FIG. 253 HUMIDITY AND PSYCHROMETRIC CHART

30.0 INCH BAROMETER

Reproduced from Carpenter on "Heating and Ventilation."] FIG. 3f.

Now the 8/10,000ths of a pound of water contained in that air does not increase in quantity, and therefore the amount of water vapour per cubic foot in the new condition of the air is

$$\frac{.0008}{1.09} = .0000732 \text{ lbs. per cubic foot.}$$

Follow, therefore, the vertical line down until it reaches the horizontal line representing .00073 lb. per cubic foot. This is almost on the curve representing 15%, which therefore represents the state of saturation of the air.

Specific Heat of Air and Water Vapour.

The values which have been given for the specific heat of air hitherto, have assumed that the air is dry. When the air is not dry the specific heat is altered very considerably, owing to the influence of the moisture in it. If there is a mass of air, say one pound, consisting of, say, 98% by weight of air and 2% of moisture, the amount of heat necessary to raise the mass through 1° is that necessary to raise the air alone + that necessary to raise the water vapour.

The specific heat of air is 0.238, so that the total quantity of heat necessary to raise the air through 1° is

$$.98 \times 0.238 = 0.233 \text{ B.T.U.}$$

per pound of mixture.

And as the specific heat of water vapour is 0.475, the quantity of heat necessary to raise the vapour through 1° is

$$0.02 \times 0.475 = 0.0095 \text{ B.T.U.}$$

per pound of mixture.

Total heat

$$0.233 + 0.0095 = 0.2425 \text{ B.T.U. per lb.}$$

from which figure the calculation can be easily made.

The change is comparatively unimportant when the amount of vapour in the air is not altered. The case is quite otherwise, however, if the amount of vapour present in the air is increased or diminished. If, for instance, the quantity of moisture in a given quantity of air has to be increased by x pounds, and if the x pounds of vapour must first be evaporated from x pounds of water at t°. Here the amount of heat necessary to perform the evaporation is considerable, amounting to about

$$x \times 1{,}078 \text{ B.T.U. per pound,}$$

the exact figures varying with the temperature.

Any problem, therefore, in which the quantity of vapour in the air has to be increased can be worked out from the diagrams given.

For instance, suppose 1,000 cubic feet of air at 40° and 76% of saturation is to be heated to 70° having 50% of saturation. The diagram shows that the amount of water in the air in its original temperature is only about 3/10,000ths of a pound per cubic foot, or .3 lb. per 1,000 cubic feet. When the temperature is raised to 70° the volume of the 1,000 cubic feet becomes 1,056 cubic feet, and the amount of vapour in the air must be about 5.75/10,000 of a pound per cubic foot, so that the total amount of water vapour in the air then must be

$$.000575 \times 1{,}056 = 0.61 \text{ pounds.}$$

The difference between these two figures is 0.31 lbs., and to evaporate this quantity of moisture at 40° F. would absorb

$$0.31 \times 1{,}086 = 336 \text{ B.T.U.}$$

To raise the temperature of 0.61 lb. of vapour from 40° to 70° would absorb

$$0.61 \times 0.475 \times 30 = 8.65 \text{ B.T.U.}$$

The quantity of heat necessary to raise the temperature of the air would be

$$79.4 \times 0.238 \times 30 = 567 \text{ B.T.U.}$$

and the apparent value of the specific heat would be

$$\frac{336 + 567 + 8.65}{79.7 \times 30} = 0.385$$

It is obvious that it makes no considerable difference to the calculation whether the moisture is calculated as being evaporated at 40° F., and subsequently raised in temperature to 70° F., or whether it is calculated as if the *water* had been first raised in temperature to 70° and then evaporated at that temperature, for the initial and final states of the vapour are the same in each case, and therefore the total absorption of heat must be the same by whatever means the evaporation was effected, neglecting the small difference in the work done.

CHAPTER V.

ON THE FLOW OF FLUIDS.

LIKE all other bodies, a fluid will only change its state of rest or motion when a force acts on it. The nature of this force in the case of a fluid must be a *pressure* equally distributed throughout the fluid. In the case of a solid hard body, of definite shape, a single force acting on any part will change the motion of the entire body, because owing to the coherence of the particles the force is transmitted by contiguous part *to every particle in the entire body*. As every particle of every body is subject to the same law, a change of motion of any solid proves the existence of a force acting simultaneously on every particle in it.

In the case of a fluid the position of any one particle is not fixed relatively to others. To transmit the force to *each particle* a pressure must be applied throughout the fluid.

Fluids have, of course, the property in common with solid bodies that they obey Newton's laws of motion.

If a *continuous* stream of fluid is started from rest the quantity of momentum generated in a unit of time must be proportional to the total force continuously impressed upon the stream. Further, the kinetic energy generated in the moving fluid in any interval of time must be equal to the work done on it during the same interval.

The bearing of these laws on the motion of fluids will be clear from a consideration of the state of things when any fluid contained in a closed vessel is allowed to escape into another vessel at a lower pressure. The laws governing the flow are substantially the same for all fluids. We will therefore consider the simplest case, namely, that of water in a tank having vertical sides.

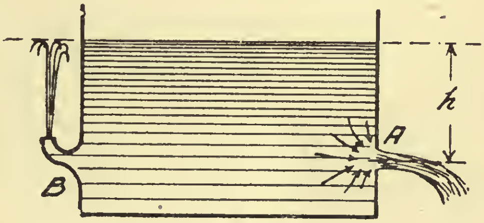


FIG. 4

Suppose Fig. 4 to represent such a tank full of water, having a small round hole in the side of the tank at a point A. The pressure at the point

A is due to the weight of a column of water, whose height, suppose, is h feet, pressing the fluid through the hole A. For simplicity, suppose the tank to have s square feet of horizontal section, and that during a certain time, t seconds, a small mass of fluid, weighing w pounds, escapes from the hole. It will evidently be squirted out with a certain velocity, which will be denoted by v feet per second.

After this small mass of fluid has left the tank the law of the conservation of energy demands that the energy in the entire system must be the same as it was originally, for energy can never be destroyed. The only form in which additional energy is manifested is in the increased kinetic energy of the moving fluid, due to the velocity at which it has been squirted through the hole.

The energy which has been lost, on the other hand, is derived from the potential energy of the mass of water originally *above* the escaped fluid, which has fallen a short distance owing to the action of gravity on it. Excluding a very minute amount of friction which will have been caused by this movement (resulting in the degeneration of a certain minute quantity of energy into heat) these two quantities of energy must be the same.

The loss of potential energy in foot-pounds is the weight of the superincumbent mass of water in pounds multiplied by the distance in feet through which it has fallen. If the weight of a unit volume (one cubic foot) of water is D pounds, the volume of the small quantity of water w pounds which has escaped is the fraction w/D of a cubic foot. Since therefore the tank is s square feet in section, $\frac{w}{D\ s}$ must represent *the distance* through which the superincumbent mass of water, whose height is h feet, whose volume is therefore $h\ s$ cubic feet, and whose weight is $h\ s\ D$ pounds, has fallen, in pressing w pounds of water out of the hole. Thus the work which has been done on the small mass of water w must be

$$\overbrace{h\,D\,s}^{\text{pounds}} \quad \times . \quad \overbrace{\frac{w}{D\,s}}^{\text{distance}} = h \times w \text{ foot-pounds.}$$

This must be equal to the kinetic energy with which the mass of water w left the tank. This kinetic energy is represented by

$$\frac{w\,v^2}{2\,g} \text{ foot-pounds (see equation 6)}$$

so that

$$h\,w = \frac{w\,v^2}{2\,g}$$

or

$$v^2 = 2\,g\,h$$

The same equation (see equation 5 on page 4) gives the velocity attained by a body falling freely through h feet. The velocity with which the water leaves the tank is therefore exactly the same as if it had fallen from the level of the surface of the water as far as the hole. Thus, if the squirt is directed upwards, as at B, the water will be projected as high as the water level before it begins to fall.

The very minute amount of friction, if taken into account, will slightly reduce this velocity. This reduction will not be more than 1% or 2%, and for our present purpose may be neglected.

The quantity of water flowing out of the hole per second will not be (as might be supposed)

$$= A \times v \text{ cubic feet per second,}$$

or the full area of the opening multiplied by the velocity, because when the hole is opened all the water round it will be trying to get out at the same time, as indicated by the arrows in Fig. 4. The rush of these various streams of water towards the same point will obviously produce the result that the individual streams will get in the way of one another. The result will be that at a short distance from the opening, the issuing stream will be smaller in sectional area than the actual opening.

Some idea of the mechanical reason for this reduction of area in the stream may be obtained by considering the matter in another aspect.

The law relating to the amount of momentum generated in a unit of time must also be obeyed as well as the law relating to the energy of the stream.

The force in poundals acting on the outflowing water, resulting in the production of the stream, must be equal to the number of units of momentum produced in unit of time. This law is as fundamentally a law of nature as that of the conservation of energy.

The difficulty of making an exact determination of the resulting velocity by calculation on the basis of this law of momentum is due to the difficulty of determining what is the exact amount of force which is acting on the water resulting in the production of the outflowing stream.

At first sight it might be supposed that the force in question is that due to the action of the pressure $h\ D$ pounds per square foot over the area A of the hole.

Assuming that this is the actual value of the force pressing the stream out of the hole, we should have

$$\text{Total force} = h\ D\ A\ g \textit{ poundals.}$$

If the mean velocity of the water when actually *flowing through the hole* is represented by V feet per second, the amount of water per second must be $A\ V$ cubic feet, weighing $A\ V\ D$ pounds.

The amount of momentum generated in one second is therefore

$$A\ V\ D \times V = A\ D\ V^2.$$

Hence on the above assumption we should have the equation

$$g\,h\,A\,D = A\,D\,V^2, \text{ or } g\,h = V^2, \text{ or } V = \sqrt{g\,h}$$

If a is the area of the reduced section where the velocity of outflow v is the greatest, it is clear that since the amount of water flowing through the hole at the velocity V must be the same as the amount of water flowing through the reduced section at the increased velocity v,

$$A\,V = a\,v$$

Therefore

$$\frac{a}{A} = \frac{V}{v} = \frac{\sqrt{g\,h}}{\sqrt{2\,g\,h}} = \frac{1}{\sqrt{2}} = \frac{1}{1.415} = 0.71$$

Accurate experimental measurements, however, show that the ratio is somewhat smaller than this when the edges of the hole are made perfectly sharp, so that the stream issues in a clean cut shape, free from dribbles and broken water, the value being about 0.61. The reason for the discrepancy is to be sought in the properties of liquid. The subject is too mathematical to be treated fully in the present volume.

It may however be explained that the inter-action between the molecules of water results, among other effects, in the production of what is called a "surface tension," whenever a free surface of water is exposed to the air. The forces in play produce the same effect as though the free surface exposed to the air were covered by a thin, tightly-drawn film or skin, which tends to contract or draw together.

The issuing stream of water may be imagined enclosed in such a skin. It is obvious that the tension of this imaginary skin will tend still further to contract the area of the issuing stream more than it would be contracted if no such skin existed.

At the same time, the existence of this tension or skin, since it is a purely passive force, cannot nullify the operation of the fundamental laws of mechanics, which are involved.

The velocity v must in the nature of things be equal to

$$\sqrt{2\,g\,h}$$

Also the velocity V, calculated through the *virtual* opening (but not the actual opening) of area A, must in the nature of things be equal to

$$\sqrt{g\,h}$$

The force acting on the stream, or in other words, the reaction tending to move the tank in the opposite direction to the movement of the stream, adjusts itself so that the actual force in poundals

$$= W\,V$$

where W is the weight of water flowing per second.

Expressed in another way, the effect of the surface tension or skin is the same as a virtual reduction of the area of the opening A. The virtual opening being $\frac{0.61}{0.71} = 0.86$ of the actual area A, and the pressure on it being

$$h\,D\,A' = 0.86\,h\,D\,A$$

These are deductions which follow directly from the fundamental laws of mechanics, combined with the results of experimental observations.

The opening A can be made to run full bore without any reduction of area if the sides of it, either inside or outside the tank, are shaped trumpet-wise to correspond with the natural normal shape of the issuing stream. The real effect of this device is to increase the *actual* area A. The real value of A is then the area of the *expanded end* of the trumpet and not the area of the opening where the water issues.

By experiment this reduction of area is found to be about 39% for a circular opening, so that the minimum sectional area of the issuing stream of water where the velocity is greatest will only be 61% of the actual area of the opening.

The slight difference between the pressure of the atmosphere on the surface of the water and at the level of the hole will not seriously affect the difference of pressure inside and outside the tank at the point A, because water weighs about 840 times as much as an equal volume of air at ordinary temperature. The difference in air pressure due to the weight of a column of air of this height, h, will be negligeable compared to the total pressure. We may therefore take the atmospheric pressure on the surface of the water to be the same as the back pressure tending to prevent the water from coming out of the hole.

Outflow of Gas under Pressure.

The theory of outflow of any gas under pressure through an opening is exactly the same as the above, but it is more difficult to grasp, partly because there is no visible difference between the gas inside the tank and the air outside, and partly because the surface of the gas in the tank cannot be imagined as open to the air as the surface of the water is.

Yet a further reason is that the difference of atmospheric pressure on the imaginary surface of the gas and at the hole cannot be regarded as negligeable, as it is in the case of a relatively heavy liquid. A further complication is that gas is not, like water, practically incompressible, but on the contrary, its volume changes with every minute change in the pressure. In order to see that the same fundamental laws must apply to the case of a gas as to that of a liquid, it is necessary to make some further explanations.

Obviously, it makes no difference to the rate of outflow of a gas under pressure *how* the pressure is actually produced. What determines the velocity of outflow is the *magnitude* of the pressure, irrespective of the way in which it is produced.

We may then imagine, as indicated in Fig. 5, the gas pressure inside the tank to be due to the weight of a column of an imaginary kind of incompressible fluid or gas, of the same weight per cubic foot as the actual gas, and that this imaginary gas is confined at the top of the tank by an air-tight piston of area a square feet, loaded with a weight w, which produces an intensity of pressure p lbs. per square foot on the piston (including the pressure of the atmosphere) which is exactly equal per square foot to the atmospheric or back pressure *p at the orifice.* The *surplus* of pressure at the orifice is then due to the weight of gas only.

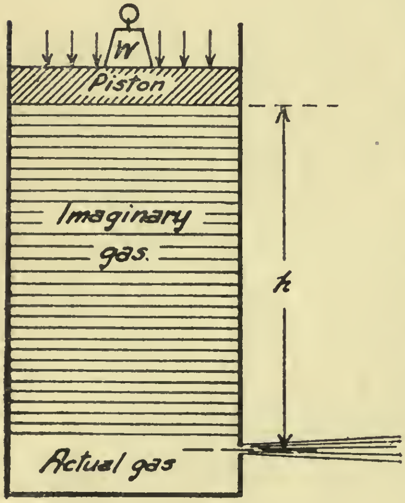


FIG. 5

Assume first that the cylinder is filled with a gas heavier than air, say carbonic acid gas, up to a level slightly higher than the hole, above this being the imaginary incompressible gas, and that the loaded piston closes in this imaginary gas at the top so as to prevent its escaping and mixing with the air. In this case it is clear that the velocity with which the carbonic acid gas will escape from the hole is the same as if it were liquid, for the only nett external work done on the escaping gas is done by the fall of the superincumbent volume of gas through w/D feet.

When any volume v of gas escapes through the hole, the piston must fall through a distance l, such that

$$l\ a = v$$

The fall of the loaded piston, however, does not do any *nett* work on the gas, because the issue of the gas through the hole does exactly the same amount of work in forcing back the atmosphere as the loaded piston does in falling.

Volume of gas forced out when piston falls l feet $= l\ a = v$, suppose.

Work done by piston in falling l feet $= p\ a \times l$ foot lbs.

But since $a\ l = v$, therefore $p\ a\ l = p\ v$.

Work done in forcing back the atmosphere by v cubic feet of air issuing against p pounds per square foot also $= p\ v$.

Whence the work done by the fall of the piston = work done by the issuing gas on the external atmosphere.

If the velocity were greater than $\sqrt{2\ g\ h}$ the additional kinetic energy acquired by the gas would be gained at no expense in energy, and the principle of the conservation of energy would be violated. Similarly, if the velocity were smaller, a certain amount of energy would vanish, which is also contrary to the law.

Any actual gas, however, if at a considerably higher pressure than the atmosphere, in escaping, will expand. In so doing the *temperature of the gas will fall.* Any *additional* velocity (*i.e.*, energy) acquired by the gas due to *expanding* into the medium into which it escapes, is acquired in virtue of the *heat* which the gas contains, and not in virtue of the energy of the superincumbent fluid (see page 37).

If the difference in pressure is small compared with the pressure of the atmosphere (as it is in all practical cases of ventilation, being at most one or two inches of water, as against the 400 or so inches of water which is the atmospheric pressure) this expansion will not be of any considerable magnitude, and is of no practical importance. In any case it is the actual velocity of the gas *at the original pressure* which is given by the formula.

For purposes of the present explanation, and in all cases of small differences of pressure, we may neglect the effect of the expansion entirely and consider that the velocity with which the carbonic acid gas escapes is due solely to the work done on it by the fall of the weight of the imaginary gas above it, as in the case of the water. In any case, for the purposes of the ventilating engineer the small error is on the side of safety.

Now the conditions will not be in any degree varied, whatever be the nature of the gas, nor whatever be the *actual* cause of the difference of pressure between the inside and outside. If the gas inside the tank is hydrogen, which is lighter than air (carbon di-oxide, of course, being heavier than air), and if a small quantity of that gas escapes, as before, through an orifice in the side of the tank, the work which is done on it in escaping must also be exactly equal to the work done by gravity on a column of imaginary incompressible gas *of the same density as the hydrogen* above it, to which the actual pressure at the orifice may be *supposed* to be due.

Note that there is no other possible source of the energy of motion acquired by the escaping gas, other than that due to the fall of the gas column above it. Therefore the velocity with which hydrogen escapes must be, exactly as before, equal to that acquired by a body in falling from the top of a column of imaginary incompressible gas, equal in density to hydrogen, whose height is sufficient to produce the excess of pressure actually found at the orifice.

This velocity will obviously be the same, however the pressure is *actually* produced. If the imaginary piston is, as before supposed, loaded with weights, such as to produce an intensity of pressure on the piston equal to the back pressure at the orifice, whatever the magnitude of the pressure may be and however it is actually produced, the reasoning will still hold good.

The general method, therefore, of calculating the velocity with which *any* fluid escapes through an opening from one pressure to a lower pressure, consists in finding what is the height of a column of fluid, of *the same density throughout* as the actual fluid, such as would produce the given excess of pressure. The velocity of outflow is the same as would be acquired by falling from that height.

EXAMPLE.—The pressure in an air flue is 1/32″ (= .0314″) of water column, and the temperature is 70°. With what velocity will air issue through a hole into the atmosphere?

Referring to Table No. V., column 16, we see that the density of air at this temperature is 833 times the standard density of water (at 62°). Therefore, the equivalent column of air at 70° is

$$H = 1/32'' \times 833 = 26.2'' = 2.18 \text{ feet}$$

and the corresponding velocity is

$$v = 8.025 \sqrt{2.18} = 11.85 \text{ feet per second.}$$

This is about the actual *velocity* which the air will acquire.

The volume of air flowing out per second will be only 61% of the theoretical quantity $A\ v$, owing to the reduction of area in the stream, as explained on page 65.

$$Q = 61/100 \times A \times 11.85 \text{ cubic feet per second.}$$

Pressure and Energy Necessary to Maintain a Current of Fluid.

It is evident from the above that if there is a continuous stream of any fluid, for instance air, A square feet in cross section, moving in a flue with a mean velocity V feet per second at a temperature t° F., at which temperature the weight of one cubic foot of the fluid is D pounds, the continuous pressure in inches of water which is necessary to push a constant supply of previously stationary air into the flue, and maintain in it a stream moving at that velocity, is given by a formula obtained as follows:—

H = height through which any body must fall to produce a velocity of V feet per sec. = height of imaginary incompressible air column =

$$\frac{V^2}{2\,g} = \frac{V^2}{64.4}$$

(The values of this expression are calculated out and given on Table IX., column 2, for different velocities.)

Weight of $\frac{V^2}{2\,g}$ cubic feet of fluid $= \frac{V^2\,D}{2\,g}$ pounds = required pressure in pounds per square foot.

(This pressure is, of course, quite independent of the pressure necessary to overcome friction of any tube or flue through which the fluid may pass. The latter must be separately calculated. See below.)

As 1 cubic foot of water at standard temperature, 62° F., weighs 62.4 lbs.

62.4 lbs. per square foot = 12″ of water column.

Dividing each side of this equation by 62.4 we have

$$1 \text{ pound per square foot} = \frac{12}{62.4} = 0.1925 \text{ inches of water.}$$

Pressure in inches of water =

$$\frac{V^2\,D}{2\,g} \times 0.1925 = \frac{0.1925}{64.4}\,V^2\,D = .003 \times V^2\,D.$$

The minimum supply of *energy* necessary per second in order to start and maintain such a stream of fluid is that necessary to give $(V \times A)$ cubic feet of fluid, each weighing D pounds (total weight $= D\,A\,V$ pounds) a velocity V feet per second.

Minimum energy expended per second

$$= \frac{(D\,A\,V)\,V^2}{2\,g} = \frac{D\,A\,V^3}{2\,g} \text{ foot-pounds.}$$

This is, of course, quite a different thing from the pressure.

It has been pointed out that the pressure necessary to start and maintain a velocity in a given body of fluid is not the same thing as the energy. It is important to know the horse-power theoretically necessary to deliver a certain quantity of fluid, either at a certain velocity or against a certain pressure.

When any given mass of fluid (or any other body, the nature of the body makes no difference) weighing W pounds is moving at a certain velocity, v feet per second, the kinetic energy it contains is

$$\frac{W\,v^2}{2\,g} \text{ foot-pounds.}$$

Therefore, if a current of air at a temperature t° F., of this velocity v, is maintained through a channel having a cross sectional area a square feet, the total amount of air started from rest in one minute is

$$60\ a\ v \text{ cubic feet.}$$

The weight of this air is

$$60\ a\ v\ D \text{ pounds}$$

where D pounds is the mass of 1 cubic foot of air at the given temperature. Since this mass has communicated to it a velocity v the theoretically minimum amount of energy which must have been applied to it in one minute is

$$\frac{60\ a\ v\ D\ v^2}{2\ g} \text{ foot-pounds.}$$

As one horse-power is 33,000 foot-pounds of work per minute, the minimum horse-power necessary, provided the whole of the propelling apparatus were of perfect efficiency, would be

$$\frac{60\ a\ D\ v^3}{66{,}000\ g} \text{ horse-power} = .0000282\ a\ D\ v^3 = \frac{a\ D\ v^3}{35{,}500}$$

If D_0 is the weight of one cubic foot of air at some standard temperature, say 0° F., at which temperature $D_0 = 0.0863$

$$D = \frac{0.0863 \times 461}{461 + t} = \frac{39.8}{461 + t}$$

whence

$$\text{Minimum theoretical horse-power} = \frac{.0000282 \times 39.8 \times a\ v^3}{461 + t} = \frac{.00113\ a\ v^3}{461 + t}$$

In practice, of course, if the air is delivered by a fan the amount of horse-power necessary is considerably in excess of this, as a fan is not a machine of perfect efficiency. The efficiency of the fan or other propelling appliance is the ratio of the theoretical work done in moving the air to the actual power which is applied to the fan.

EXAMPLE.—Find the minimum horse-power necessary to maintain a flow of air of 850,000 cubic feet per hour at 80° F. through a hole 3 feet in diameter. Here

$$v = \frac{850{,}000}{3{,}600 \times 7.05} = 33.5 \text{ feet per second.}$$

$$\text{Horse-power} = \frac{.00113 \times 7.05 \times (33.5)^3}{541} = \frac{.00113 \times 7.05 \times 37{,}500}{541} = 0.55 \text{ H.P.}$$

The actual horse-power would, of course, be greatly in excess of this, depending on the efficiency of the fan and the friction in the tube.

If this volume $a\ v$ of fluid were not delivered into the free atmosphere, as in the first case, but were delivered at a velocity v against a head pressure h_1 feet of fluid column, which pressure is subsequently destroyed, say, in overcoming the friction of flues, it would make no difference to the power which must be supplied to the fluid how the pressure generated is subsequently utilised. We may therefore imagine, if we please, that the pressure is actually utilised in producing a velocity (as in the first case) in the stream of fluid. As the *actual* velocity of the fluid, as delivered against the pressure, is v, we can convert this into an additional pressure

$$h_2 = \left(\frac{v_1{}^2}{2\,g} \right)$$

and add the two pressures together, so that the actual work which is done on the fluid is equal to the work which would be done in lifting the same volume at no velocity up to the top of a column of $(h_1 + h_2)$.

This being so the energy in the mass of fluid is equal to

$$\frac{60\,(a\,v\,D_0)}{1 + a\,t^\circ} \times \left(h_1 + \frac{v_1{}^2}{2\,g}\right)$$

and the horse-power required is

$$\frac{0.00182}{1 + a\,t^\circ}\,a\,D_0\,v\,\left(h_1 + \frac{v_1{}^2}{2\,g}\right)$$

EXAMPLE.—Find the theoretically minimum horse-power necessary to deliver 5,500,000 cubic feet of air per hour at 50° into a flue 7 feet × 5 feet against a pressure of ¾ inch of water.

$$\text{Here } v_1 = \frac{5{,}500{,}000}{3{,}600 \times 35} = 43.5 \text{ feet per sec. actual velocity.}$$

$$h_2 = \frac{(43.5)^2}{64.4} = 29.3 \text{ feet head of air column.}$$

$$h_1 = 0.75 \times \frac{62.4}{.078} \times \frac{1}{12} = 50 \text{ feet head of air column.}$$

$$(h_1 + h_2) = 79.3 \text{ feet.}$$

Hence horse-power =

$$\frac{0.00182 \times 35 \times 0.0863 \times 43.5 \times 50}{1.108} = 10.8 \text{ H.P.}$$

In order that these calculations in any given case may be easily carried out, the Table No. IX.*a* is given, column 2, calculated from the ordinary formula, giving the velocity acquired by any body in falling through different distances, and columns 3 to 8 the corresponding pressure in inches of water necessary to discharge air at different temperatures at the given velocity.

It is important that the manner in which these tables are calculated should be thoroughly understood. To this end the question of *units* must receive close attention.

In obtaining numerical results from any formula, the chief difficulty is to express correctly the units in which the result is expressed.

A careful study of the section devoted to units is recommended.

Flow of Fluids through Conduits and Pipes.

This is quite a different proposition from the preceding one. A fluid will not flow from one point of a conduit A to another point B, unless there is either a greater pressure at A than at B, or sufficient kinetic energy in the fluid to force it along against a difference of pressure. If a moving mass of fluid has already been propelled along the conduit at a certain velocity, the moving mass will continue to move until some resistance stops it, by acting upon it for such a time that the product of the total resisting force in poundals into the time during which it acts (seconds) is numerically equal to the momentum of the mass of fluid.

If the moving mass is an isolated volume of fluid, it comes to rest almost immediately, because (not to mention the resistance due to rubbing the side of the conduit) before it can pass along the conduit at all it has to push out in front of it the rest of the fluid in the conduit and to draw fluid in after it to fill up the space behind it, which its own motion would otherwise leave vacant. If there is to be maintained a *continuous* stream of fluid through a conduit there must be a *continuous* supply of power behind it to push it along, or in front of it to suck it along—really the same thing.

No fluid contains energy merely in virtue of its pressure. The pressure behind it is not in itself energy. The energy necessary to drive a column of fluid along a conduit is contained in the superincumbent mass, the fall of which (or its equivalent in pump, fan, or other power) supplies the energy to the mass of moving fluid. The continuance or maintenance of the pressure is only the *evidence* of the existence of the energy, and the pressure itself is the link through which that energy is supplied to the fluid, just as the stress in the draw-bar of a locomotive, though not in itself energy, is one of the links by which the energy of the steam is transmitted to the train. Numerically the value of the pressure or head in suitable units is the same as that of the energy which is communicated to every pound of the fluid by the mass of fluid above it.

Transformations of Energy in a Conduit.

If a uniform stream of fluid moves along a conduit in which the sectional area varies, the same weight of fluid must pass any two sections of the conduit in the same interval of time, otherwise there would be either an accumulation or an abstraction of a certain quantity of fluid in or from the space between the two sections.

Thus, if the areas at two points are a_1 and a_2 and the velocities v_1 v_2 respectively, we must have

$$a_1 \, v_1 = a_2 \, v_2.$$

But the energy in unit mass of the fluid varies as the square of the velocity, so there must be continuous changes in the kinetic energy per pound whenever a stream of fluid passes through a variable channel, and the question arises, since energy cannot be created nor destroyed, where does the access of energy come from when the velocity increases, or where does it disappear to when the velocity diminishes?

The question is further complicated by the possibility of an alteration in the level in the channel and pipes. When a pound of fluid flows through a pipe to a point 1 foot lower down it certainly loses 1 foot-pound of potential energy. Where does this energy go to? If it flows up one foot it must gain 1 foot-pound. Where does this energy come from?

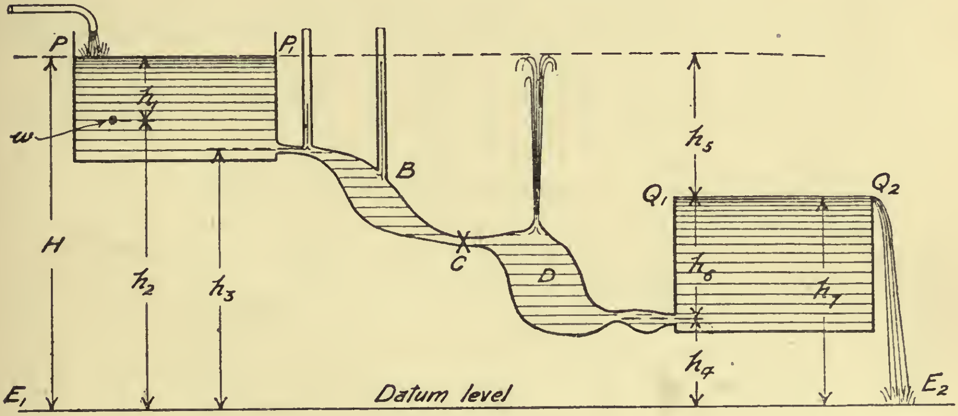


FIG. 6

The heating engineer has to deal with streams of fluids in complicated systems of conduits, and in which the temperature, the density, the velocity, and the level, all vary between wide limits. In order to gain a clear appreciation of the conditions of circulation in such a system, it will be well to consider carefully the variations in pressure and energy in any system of pipes through which any fluid flows.

Consider the simple case of a tank containing water, as shown in Fig. 6. A pipe or conduit is connected to the bottom of the tank, through which water flows to another tank or reservoir at a lower level.

The principle of the conservation of energy proves that, during the flow of water, no energy can on the whole be lost. Any energy that exists in the water, due to its position, can only be transformed into some other kind of energy, and after the whole of the water has flowed out of the tank the sum total of energy existing, either as potential energy (*i.e.*, energy of position), kinetic energy (*i.e.*, energy of motion), or heat energy, must be absolutely identical with the potential energy in existence before the flow of water was started.

In order to fix the mind on definite quantities, conceive the energy is calculated above a certain datum level, which is shown on the figure as $E_1 E_2$. When the water is at rest in the tank and pipes, before the valve is opened, the total energy of the entire system is the sum of the quantities of potential energy in each separate particle.

Total energy $= \Sigma\, w\, h$, calculated above the datum level and above the then existing temperature of the water, where w is the weight of any particle, and h is its height above the datum level.

The absolute amount of potential energy contained in any given particle at any moment is proportional to its height above the datum level at that moment. The *maximum* amount of energy in any one particle is therefore that of a particle at the surface of the water in the tank, for this is the highest level in the entire system. Any of the particles in the entire system can have an equal amount of energy communicated to it, merely by circulating it to the free level of the water, but if this is done the actual source of the increase of potential energy absorbed by the particle during the process of lifting is derived from the sinking of some other particle which it displaces. The same particle may have an equal amount of kinetic energy communicated to it in another way, namely, by allowing it to squirt out through a hole into the open air with the pressure at the back of it.

It has been shown on page 64 that excluding friction, the kinetic energy which this particle will thus obtain is exactly equal to that derived from falling from the free water level, *and it derives this energy entirely from the fall of the superincumbent mass of water through a fractional distance.* This energy is called the "pressure energy" of the particle, but it must be carefully observed that the energy which the particle has, and which is called its "pressure energy," is not in fact due to its own pressure at all. It is actually due to the potential energy of the particles above it. When this so-called "pressure energy" is utilised, or converted into kinetic energy, an equivalent amount of potential energy disappears at the same time from the mass of fluid above it. *Mere pressure as such does not give any energy to any fluid, compressible or otherwise.* The pressure is merely the link or means by which the energy resident in the upper part of the fluid (and where the fluid is a gas—*in its own heat*) is conveyed to the moving particles.

If a continual stream of fluid be allowed to escape, thereby generating a continual stream of kinetic energy in the outflowing stream, the potential energy of that part of the fluid above the orifice is diminished by exactly the same amount in the same time.

If the level of the water in the tank is maintained constant by an inflowing stream of water, the potential energy of the entire system is thereby maintained constant. The escaping energy of the outflowing squirt is then really derived from, and is exactly equal to, the potential energy introduced into the system by the inflowing stream at the surface level. The mass of water between the two merely serves as a connecting link to transfer the energy from the inflowing to the outflowing stream, and only as the source of the energy, in so far as it sinks to a lower level, this loss of energy being constantly made good to the system by the inflowing stream of water.

To speak, therefore, as is sometimes done, of "the energy of the stream" in a pipe as being constant, is a fundamental mistake, unless the supply of energy into the upper part of the tank is taken account of. Further (*still assuming that the level of the water in the tank is maintained by an inflowing stream*), the energy (calculated from the datum level) of any particle of weight w, at any height h above the datum level at any given moment, consists of three parts.

(1) Its absolute potential energy (calculated from the datum level) whose value is $w\,h$.

(2) Its kinetic energy, or energy of motion at the moment, whose value is $\frac{w\,v^2}{2\,g}$ foot-pounds. This portion of its energy has been derived from the potential energy of the fluid originally above the particle. The latter has sunk to a lower level during the process of accelerating the particle, and has thereby communicated some of its own potential energy to the particle in question in the form of kinetic energy.

(3) What is called its "pressure energy," derived directly from the potential energy of the inflowing stream and does not really belong to the particle itself at all, which is the same as if the particle existed at the free level of the water, namely, $w\,h_1$ where

$$h_1 + h_2 = H.$$

This energy would suffice if it were communicated to the particle to squirt it up to the free water level, as has already been explained.

These different components of the total energy are by suitable means convertible into one another. Thus, if the sectional area of a horizontal pipe through which a stream of fluid is flowing varies gradually by expanding or contracting, the consequent decrease or increase of velocity in the pipe *is always accompanied by an increase or decrease in the pressure.* This is more fully explained later in the present chapter. Thus, if a stream of water flowing in a full pipe, whose velocity is 3 feet per second, gradually slows down, owing to a gradual enlargement of the pipe, until at another section its velocity is 2 feet per second, the actual pressure as measured by a water gauge would increase by

$$\frac{3^2}{2\,g} - \frac{2^2}{2\,g} = \frac{5}{64.4} \text{ feet, head of water.}$$

Loss of Energy through Sudden Changes in Velocity.

If, however, a pipe through which any fluid is moving with a certain velocity v, and containing therefore an amount of energy per pound weight equal to $\frac{v^2}{2\,g}$ foot-pounds is allowed to flow into a large tank or reservoir, where it is brought practically to rest, the whole of that energy is lost and converted into heat.

The stream of fluid entering the tank with the given velocity will force a cylindrical passage for itself into the still fluid, and the friction of this stream will draw into the current other standing fluid in its immediate neighbourhood. The cylindrical stream will gradually expand trumpet-wise, disturbing the standing fluid around it, and ultimately destroying by friction all the energy of the stream.

The case, however, is quite otherwise if the fluid is allowed to come to rest *gradually*, owing to gradual enlargement of the pipe, for here no local currents are formed which waste the energy of the moving fluid, but instead the gradual reduction of the velocity raises the pressure in front of the stream and communicates the increased pressure to the remainder of the fluid.

If the enlargement of the pipe is again reduced at the other end of the reservoir or tank in the same gradual way, the pressure accumulated by the gradual stopping of the stream is again utilised in starting a fresh stream without substantial loss. The energy is merely transferred from one point in the conduit to another point further on.

In order that there may be absolutely no loss, otherwise than by the inevitable friction against the sides of the channel, the inlet and outlet must of course be carefully designed, but if the stream is allowed to pass into the expanded portion without such careful design the loss of the entire energy of the moving stream must be counted upon.

In the case of a bend similar considerations hold. If the fluid is driven up into a sharp angle, and a sudden change of direction is caused by a sharp bend, not only is the whole of the kinetic energy of the fluid in one direction destroyed and a fresh stream started in a direction at right angles, but the fluid is mixed up and violent eddies are formed and the kinetic energy is rapidly converted into heat. This energy, and more energy than necessary, is thus wasted in starting the velocity again in a direction perpendicular to the original direction. This effect may be softened down to any extent by causing the change of direction to be gradual.

If a bend is made with a large radius, the loss of energy in passing round such a bend is comparatively small. Such as it is, it is caused, of course, by the fact that the fluid at the outside of the bend must move on the whole faster than that at the inside of the bend, and friction is thus caused between the various layers of fluid.

Transformations of Energy in a System of Pipes.

In order to acquire a perfectly clear conception of the distribution of energy in a moving stream it will be well to trace out the source and the destination of each part of the energy of one particle of a stream, such as is illustrated in Fig. 6.

Assume that a particle of weight, w lbs., is introduced into the tank at the surface, containing therefore potential energy $w\ H$ foot-pounds, at a temperature t°. Its total heat energy reckoned above temperature t° is therefore zero.

As the water then in the system is forced out through the pipe towards the lower tank, this particle, along with other similar ones, gradually sinks. It thereby communicates—

(1) Some of its energy to the particles in the pipe (*e.g.*, at A), which are thereby increased in velocity, and gain kinetic energy at the expense of some of the potential energy of all the particles above them.

(2) Some of the potential energy of the particle is also *transmitted by the pressure* along the whole length of the conduit, forcing the particles into the lower tank against the head or back pressure prevailing there.

Ultimately this particle parts with all of its own energy, except a quantity $w\ h_3$, by communicating it in one or other of these two ways to the particles lower down.

It now enters the pipe. It is immediately forced by the pressure above it into the pipe with a velocity v_1 feet per second, and thus acquires kinetic energy $\frac{w\ v_1{}^2}{2\ g}$ at the expense of the potential energy of particles which fell into the tank after it had itself sunk below the surface, and which are now themselves in process of gradually sinking. It retains this kinetic energy as long as its velocity is v_1. But when the pipe gradually enlarges, the velocity v_1 diminishes to v_2 at B. The particle thereby loses an amount of that part of its own kinetic energy due to its velocity

$$= w\left(\frac{v_1{}^2 - v_2{}^2}{2\ g}\right)$$

which it had previously acquired from the particles above it, as already described.

The question arises, where does this energy go to?

If a pressure gauge is fixed at the point A, and another one at the point B, it will be found that the water stands at B at a considerably higher level than it does at A. This difference of level represents the increase in pressure which has been communicated to the water, on account of the gradual retardation of the velocity of this and other particles.

But it has already been explained that mere pressure does not confer energy on any body whatever. Wherever the energy goes to, it is certainly not into the water at B.

The effect of the increased pressure, however, is to accelerate the velocity of the water at C, *in front of* the mass at B, where (assuming the sectional area at C to be the same as at A) the velocity again becomes v_1. The energy of the particle as it existed at A is thus directly transferred *to some other particle* at C. *The means by which the transference is effected is the pressure which is generated in the relatively large mass of water* B. This is the only sense in which the term "pressure energy" has any meaning.

The same process of transformation goes on at C, that is to say, the particles of water at C, moving with the velocity v_1, part with their energy, and by raising the pressure in the enlarged mass of water D, they thereby transfer their energy to a point further on. This process is continued right along the conduit.

Assume that at F the area of the section is the same as at A, the velocity of the water must also be the same, v_1. The potential energy of the particle w when it arrives there is $w\ h_4$ only, but its velocity being v_1, the *total* energy then inherent in the particle itself

$$w\ h_4 + w\ \frac{v_1{}^2}{2\ g} \text{ reckoned in foot-pounds.}$$

At this point it rushes with the velocity v_1 into the still mass of water in the lower tank, and as already explained, the whole of the kinetic energy represented by $w \frac{2\,g}{v_1{}^2}$ is then converted into heat.

Now we have not, of course, considered the question of the friction of the water in the conduit. We have assumed for the purposes of this explanation that the liquid passes through the conduit without friction. If it does so the value of the velocity v_1 must be the same as would be acquired in falling through a height h_5. The amount of energy which has been turned into heat will be $w\ h_5$ foot-pounds.

Of the original quantity of energy $w\ H$, the amount of potential energy left is then $w\ h_4$. $w\ h_5$ has been converted into heat and the remainder, $w\ h_6$, is accounted for as follows :—

The volume of the particle is $\frac{w}{D}$ cubic feet. In entering the tank it has to force its way against a pressure $h_6\ D$ pounds per square foot. The energy required to force it in is therefore

$$\frac{w}{D} \times h_6\ D = w\ h_6.$$

When therefore the particle w forces its way into the lower tank, impelled by the pressure behind it, it communicates an amount of energy represented by $w\ h_6$ *to other particles in the tank,* which are thereby forced upwards towards the level $Q_1\ Q_2$.

The energy which thus forces the particle into the lower tank against the back pressure existing at F is not at the time inherent in the particle itself, but is transmitted to it by and is part of the potential energy of the particles behind and above it.

At a previous period this particle itself, in the process of forcing *some other earlier particle* into the tank, has already delivered up the same quantity of energy as is now required to force itself into the lower tank. Its own previous expenditure of energy in the common cause is now as it were made up to it by other particles which perform the same service for it as it has itself already performed for previous particles.

These other displaced particles then overflow at the point Q_2, and in freely falling as far as the level $E_1\ E_2$ they acquire kinetic energy, whose value is $w\ h_7$ foot-pounds. Ultimately when the particle reaches the level $E_1\ E_2$ of the total amount of energy $w\ H$ originally in it, $w\ h_5$ exists as heat energy, and the remainder, or $w\ h_7$, exists as kinetic energy acquired in falling through the distance h_7. If the particle actually falls on hard ground at $E_1\ E_2$, this kinetic energy, $w\ h_7$, is also splashed away into heat.

Thus, it will be seen that none of the total energy $w\ H$ is lost by flowing through the conduit.

This balance sheet of the energy of the particle should be carefully studied, so that the student may be perfectly familiar with the very interesting transformations of energy which take place in a variable stream of fluid, and so that he may not be misled by the incorrect expression "pressure energy," which is so frequently used.

Loss of Head through Friction in a Pipe or Flue.

There is a certain amount of pressure lost for every foot of conduit along which a stream of fluid travels, owing to friction between the fluid and the sides of the conduit, and internal friction in the fluid itself.

When air or any other fluid passes through a pipe, it does not flow as would a *solid* piston-like mass of unchangeable form. The layer of fluid in contact with the walls of the flue tends to adhere to them. That layer (which we may imagine for present purposes, to fix our ideas, as cylindrical and, say, ⅛″ in thickness) adheres more or less to the walls of the flue. The next layer, proceeding inwards, which we may also imagine ⅛″ in thickness, tends to adhere to the outer layer No. 1. The next layer, No. 3, tends to adhere to No. 2, and so on until we get to the centre of the conduit. It will thus be clear that the fluid at the *centre* of the conduit must be moving faster than that at the sides, and that the velocity gradually increases from the outside to the centre. The difference in velocity depends on the " viscosity " of the fluid.

The actual motion of the fluid may be most easily imagined by conceiving that it consists of a large number of concentric cylindrical layers or rolls of blotting paper. If such a cylindrical mass were impelled along the tube by a uniform pressure at one end, the outer layer of blotting paper would adhere to the wall of the pipe, the core of the roll would move forward, and the intermediate cylinders would rub against one another, and would, in so doing, produce little rolls of blotting paper which would be rubbed away from the substance of each layer, so that in a short time the originally flat end of the cylindrical mass of blotting paper would form a sort of cone, and between each layer there would be minute rolls of the substance of the paper formed by the friction of two successive layers on one another.

This is exactly what happens when any fluid flows through a tube, and this passive force by which some of the energy is destroyed or wasted in the rubbing of the layers over one another, and in the formation of these minute rolls is what is called the "frictional resistance " of the tube.

Circulating Pressure in a Pipe.

It has been stated as an obvious fact that water cannot flow along a pipe unless the pressure at the beginning of it is greater than the pressure at the end of it, by the amount of the resistance which is due to friction of the water in the pipes. As this is an important doctrine which must be clearly understood by the heating engineer, it is necessary to explain exactly what it means.

Without some explanation the statement clearly does not hold good. For instance, it is a regular phenomenon that water will flow down a pipe, although the absolute pressure at the bottom of the pipe must be greater than that at the top. What will in future be called the " circulating pressure " in this case, as in all others, must be taken to mean *the difference in actual pressure at the two ends of the pipe, less the pressure caused by* *the weight of a column of water of the same mean density as that in the pipe, whose height is the difference in level between the upper section and the lower section.*

In order to determine, therefore, whether water will flow up or down a pipe between two points A and B, Fig. 7, the water being at a mean temperature *t*°, it is necessary to ascertain the actual pressure at A and B in pounds per square foot, and to take the difference between those pressures. Subtract from the difference the pressure due to a water column of the same mean density as the actual column, $H \times D_t$, and whose height (H) is equal to the vertical height from B to A.

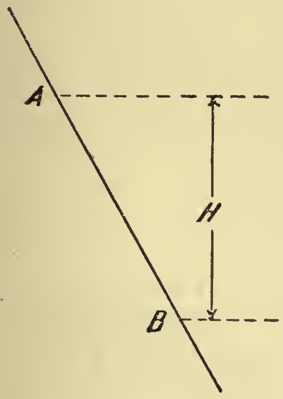


FIG. 7.

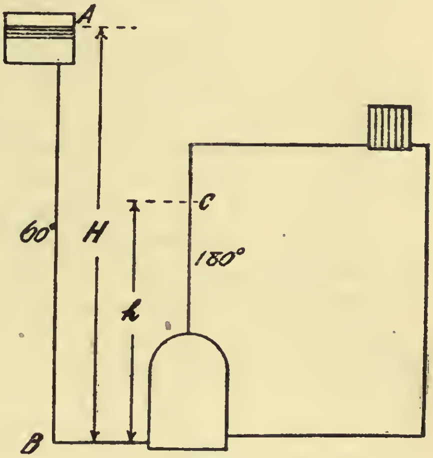


FIG. 8.

If the result is zero the water will remain at rest, if positive, water will flow up the pipe, and if negative, down. In all cases in this book this difference is the pressure spoken of as the " circulating pressure " between the two points A and B. If this is clearly understood we can speak of the total circulating pressure acting throughout any circuit without reference to the actual level or absolute pressure at any particular point, for the absolute static pressure will be eliminated by the process of calculation.

To ascertain by calculation the absolute static pressure at any point will involve some complexity of figures. For instance, in the system shown on the sketch Fig. 8, assume that the temperature in the main feed or expansion pipe is 60° F., the absolute pressure at the bottom of the boiler will be that due to a water column, whose height H is $A B$ at, say, 60° F. The pressure at the point C will be this maximum pressure at B, less the weight of a water column at the *temperature of the riser, say,* 180°, equal to the height h of that riser up to the point C.

If the density D_t of the water at each temperature is expressed in pounds weight per cubic foot, the pressure (above atmospheric) due to any column (in pounds per square foot) is the density of the water multiplied by the height of the column. This should be reckoned for each section of pipe of which the total head is composed, and the results added together, or subtracted, as the case may be.

In the above example the actual pressure at C when the water is at rest is

$$H D_{60} - h D_{180}.$$

Magnitude of the Resistance.

The fundamental rules by which the magnitude of these resistances is calculated are the same for all fluids. Almost the only fluids with which the heating engineer has to deal are air, water and steam.

Experiments have been made on the flow of all these fluids through pipes and channels, and the law governing the magnitude of the resistance has been found to be of the same general form for each.

The most convenient method of measuring the resistance is in terms of the height of a column of the same fluid which is necessary to produce the pressure required to overcome the resistances.

Referring back to page 70 it will be seen that the pressure necessary to start from rest a continuous stream of fluid, with velocity v, is given as the height (h) of a column of the same fluid which is necessary to produce the pressure where $h = \frac{v^2}{2g}$

The numerical values are shown on Table IX. and on Figs. 8*a* and 8*b*. Alternatively, this same figure represents the number of foot-pounds of kinetic energy resident in a pound of fluid moving at the given velocity v.

In every case in which any fluid travels along a flue of uniform cross section, there is from point to point a continuous conversion of a portion of the *total energy* of the system into heat, due solely to friction within the fluid. This conversion is manifested at the successive points as a continuous loss of pressure.

Loss of pressure is not in itself necessarily a loss of energy. In this case it is only a manifestation or evidence of that loss.

It is due to an entirely different cause from the loss of pressure by conversion into kinetic energy (to use the same simile as has been used before), just as the part of the force which a locomotive exerts on a train and which is necessary to overcome the friction of the axles in their bearings, of the wheels on the rails, and other frictional forces, and so *to keep up* the speed, is an entirely different thing from the force which has been and is being exerted to accelerate the motion and *get up* the speed of the train.

The sum total of the pressure necessary to start and maintain the current in the flue is the sum of the two sections. We have already calculated the magnitude of the recoverable portion of the energy manifested as pressure necessary to start the fluid off with the required velocity. The method of calculating the magnitude of the loss due to the frictional resistance, which is irrecoverable as mechanical energy, will now be explained.

Frictional Resistance.

It is found by experiment that the loss of energy or reduction of pressure caused by the flow of any fluid at a certain velocity past any obstruction, or through a given length of channel, or through an opening, is in all cases roughly *proportional to the square of the velocity,* when the velocity is within the limits with which the heating and ventilating engineer has to do.

Now the pressure *necessary to start and maintain the current,* which has already been dealt with, is also proportional to the square of the velocity. It is therefore extremely convenient to regard the resistance to the flow of fluid past any obstacle as being proportional to the pressure required to generate the current, which latter is a factor we can easily calculate and give in tables.

There is, however, no fundamental connection or relation between these two components. It is more or less of a scientific accident that one is exactly and necessarily proportional to the square of the velocity, and the other approximately proportional to the same thing. Use is made of this accident in order to simplify the necessary formulæ.

The pressure necessary to maintain a current of fluid through a conduit and overcome all the resistance is then composed of three parts.

(1) That necessary to start the current and maintain it.

(2) That necessary to overcome the friction in the straight lengths of conduit.

(3) That necessary to force the fluid past any single resistance, such as a valve, bend, or other obstruction to the freedom of flow.

The magnitude of the second and third terms is therefore the value of the first term multiplied by a factor, which factor varies with the nature and size of the pipe or obstacle.

The total resistance for any run of pipe of the same size throughout therefore

$$= \frac{v^2}{2g} + M\frac{v^2}{2g} + N\frac{v^2}{2g} \text{ in feet of fluid column,}$$

where M represents an unknown co-efficient corresponding to the straight length of pipes, and N a similar unknown co-efficient corresponding to the sum of the individual resistances.

Value of M.

It has been explained above that the force necessary to move any fluid along a conduit is due to the fact that the layer of the fluid in close contact with the walls of the conduit clings to those walls, and that the movement of a body of the fluid cannot be effected without a sort of shearing action taking place between the fluid clinging to the walls and the rest of the fluid. It is therefore reasonable to suppose that the total *force* necessary to move the fluid through a pipe would be proportional to the area of fluid which has to be sheared through. This is found to be borne out by experiment. The force required does not in the least depend on the absolute value of the pressure. It requires no greater *difference* of pressure at the two ends of the pipe to propel water at 1,000 pounds per square inch than at 5 pounds.

The area sheared through is obviously that of the interior wall surface of the channel, against which the fluid rubs. Now, as explained earlier in the present chapter, this total force is naturally distributed in the form of a uniform *pressure* over the whole cross sectional area of the channel. The total *force* in pounds consists therefore of the difference in *pressure* (in pounds per square foot) at the two ends of the channel multiplied by the number of square feet in the cross sectional area, that is

$$\text{Total } force \text{ on fluid column} = p \times a$$

where p is the difference of *pressure* in pounds per square foot, and a is the sectional area of the flue in square feet.

If the pressure is expressed as h feet of fluid column instead of as p pounds per square foot we have $h\,D = p$.

Therefore, since the force is proportional to the square of the velocity and to the total area rubbed against, we have

$$p \times a = \rho \times l \times c \times \frac{v^2 D}{2\,g} \text{ pounds}$$

where ρ is the co-efficient, the value of which has to be determined, l is the length and c the circumference of the conduit, both in feet. Expressed in words, ρ is the proportion between the force necessary to shear through one square foot of rubbing surface at the given velocity and the force required to start the stream from rest. The $2\,g$ is introduced for convenience only, in order to express the pressure required to overcome friction as a multiple of the pressure necessary to start the stream from rest.

Now it is very desirable to obtain the power necessary to force the fluid along the conduit in terms of the difference in *intensity of pressure only*, and not of the *total force* on the fluid column. This is done by dividing each side of the equation by a, the area of the conduit, and the formula becomes

$$p = \rho \frac{l\,c}{a} \times \frac{v^2 D}{2\,g} \text{ pounds per square foot,}$$

or in feet of fluid column

$$p = \rho \frac{l\,c}{a} \times \frac{v^2}{2\,g}$$

The numerical value for ρ depends on the units in which p is expressed. For convenience *p is always expressed in feet or inches of fluid column.* Thus, where air is flowing in a flue consisting of a rectangular channel, whose sides are x and y, the total resistance

$$= \rho \frac{2\,l\,(x + y)}{x\,y} \frac{v^2}{2\,g} \text{ in feet of air column of the same temperature.}$$

Where the conduit is a round pipe, the formula becomes

$$\rho \frac{l\,\pi\,d}{\frac{\pi\,d^2}{4}} \times \frac{v^2}{2\,g} = 4\,\rho \frac{l}{d} \times \frac{v^2}{2\,g}$$

The value of ρ generally varies with the circumference and with the velocity of the current, and with the character of the fluid. Its magnitude for different kinds of fluid will be discussed later.

Value of N.

Each individual resistance has its own co-efficient. The co-efficient in this case is generally indicated by ζ. No general formula can be given for the value of ζ, the values are determined purely by experiment for each class of resistance.

Expressed in words ζ is the proportion of the kinetic energy in one pound of the fluid which is required to force one pound of the fluid past the obstacle at the same velocity; or in other words, the ratio between the head necessary to force the stream past the obstacle and that required to start the stream from rest.

There are, of course, an infinite number of shapes which any kind of resistance may take, and it is impossible in practice to give a separate value of ζ for each separate shape, so the values have to be given as a rough approximation.

The values actually given in each case are liberal and will be found to be on the right side for most kinds of resistance.

We have then the general formula for the resistance in feet of fluid column, when any kind of fluid is started from rest and flows through any conduit.

Total resistance =

$$\underset{\text{Starting.}}{\frac{v^2}{2g}} + \underset{\text{Straight part.}}{\frac{\rho\, l\, c}{a}\,\frac{v^2}{2g}} + \underset{\text{Single resistances.}}{\Sigma\,\zeta\,\frac{v^2}{2g}}$$

in feet of fluid column of same density as the actual fluid.

These formulæ have now to be applied to the different kinds of fluid with which the heating engineer has to deal.

Flow of Air in Flues.

In the case of a flue of moderate size, along which a current of air is flowing, it is easy to detect the difference in velocity at different points in the section, by placing an anemometer at different distances from the axis. It would be found in every case that the velocity at the centre of the flue is greater than at the sides, except in cases where there are obstructions or other reasons for the formation of local currents of greater or less velocity within the tube itself. Such cases, however, cannot be taken account of, and we must assume that the motion of the air throughout the whole flue is uniform and equal to the mean, or that

V in cubic feet per hour =

$$\underset{\text{ft. sec.}}{v} \times \underset{\text{ft.}^2}{a} \times \underset{\text{secs. per hour.}}{3{,}600}$$

As before, in calculating the total loss of pressure which will be caused in a flue by the passage of air along it, there are three main sections to take account of.

(1) The loss due to the maintenance of the current.
(2) The loss due to friction in the straight part of the flue.
(3) The losses due to the resistance.

The general formula is as before.

Total resistance in feet of air column

$$= p = \frac{v^2}{2g} + \frac{\rho\, l\, c}{a}\,\frac{v^2}{2g} + \Sigma\,\zeta\,\frac{v^2}{2g}$$

Value of ρ.

The values of ρ differ according as the flue is constructed, with very smooth walls, such as metal flues, or rough builders work.

Very extensive experiments have been made by Rietschel, and the value of ρ obtained from these experiments is, reduced to English measure, as follows :—

For very smooth channels, such as metal flues,

$$\rho = 0.00309 + \frac{0.00687}{v} + \frac{0.001105}{c} + \frac{0.00945}{c\,v}$$

in which v is the velocity of the air in feet per second, and c is the circumference of the pipe in feet.

These values of ρ have been calculated out by the Author, converted into British units, and exhibited on curves, Fig. 8*a*.

It will be seen that the values of ρ vary considerably according both to the velocity and size of flue.

In order that calculations of the volume of air delivered may be rapidly made, the diagrams in Chapter XXXI. have been calculated from Rietschel's formula by the Author. The diagrams show the loss of head in feet of air column for different deliveries per hour, through different sizes of round flue, 10 feet long in each case.

For clean built flues, Rietschel gives values, given in the Appendix, for built flues above 20″ circumference. Below this value the co-efficient is given according to Rietschel by the equation

$$\rho = 0.0065 + \frac{0.0604}{2.54c - 48}$$

This formula is not in conformity with the given values for smooth channels, doubtless because exact values for ρ for rough channels cannot be determined on account of the great variations in the roughness of the walls, which has a very large effect when the diameter is small.

Rectangular Flues.

In order that the values shown on the diagram may be generally applicable to rectangular pipes as well as round pipes, it is well to determine the dimensions of a rectangular pipe having the same carrying capacity as a round pipe. By this is meant that when the *volume* of air delivered per hour by the rectangular pipe is the same as by the round pipe, the head lost per 10 foot run must be the same.

It is clear that the velocity of the air must be slightly different in the case of the square pipe from what it is in the case of the round, because no other cross section can have exactly the same area, and at the same time exactly the same perimeter as the circle. If, therefore, the friction is to be the same, and *also* the delivery, the velocity must be different.

(1) To ascertain the side of a square flue having the same carrying capacity as a round flue.

Let v_1 be the velocity of the air through the round flue, and v_2 the corresponding velocity through the square flue. Let d be the diameter of the round flue, and x the length of the side of an equivalent square flue. Then it is clear that as the total quantity of air delivered per second is the same in the two cases

$$\frac{\pi}{4} d^2 \times v_1 = x^2 \times v_2$$

or

$$v_2 = \frac{\pi\, d^2}{4\, x^2} v_1$$

Also, since the frictional head lost in the two cases is the same, we must have

$$\frac{4\, \rho_1\, l}{d} \times \frac{v_1{}^2}{2\, g} = \frac{4\, \rho_2\, x\, l}{x^2} \times \frac{v_2{}^2}{2\, g}$$

or cancelling out common terms

$$\frac{\rho_1}{d} v_1{}^2 = \frac{\rho_2}{x} v_2{}^2$$

Hence, substituting for v_2 its value in terms of v_1, derived from the upper equation,

$$\frac{\rho_1}{d} v_1{}^2 = \frac{\rho_2}{x} \left(\frac{\pi}{4}\right)^2 \frac{d^4}{x^4} v_1{}^2$$

whence

$$4^2\, \rho_1\, x^5 = \pi^2\, \rho_2\, d^5$$

$$x = \sqrt[5]{\frac{\pi^2}{4^2} \frac{\rho_2}{\rho_1}}\, d$$

Now referring to the diagram showing the value of ρ, it will be seen that a slight difference in velocity and perimeter will make a comparatively small difference in the value of ρ. In order, therefore, to avoid complications, it will be sufficiently accurate to assume that the value of

$$\rho_1 = \rho_2$$

whence we obtain

$$x = \sqrt[5]{.617}\, d$$

which gives

$$x = 0.91\, d$$

It is thus clear that in all cases the curves given on the diagram may be taken to be the curves for a square flue, having a side = 0.91 of the given diameter.

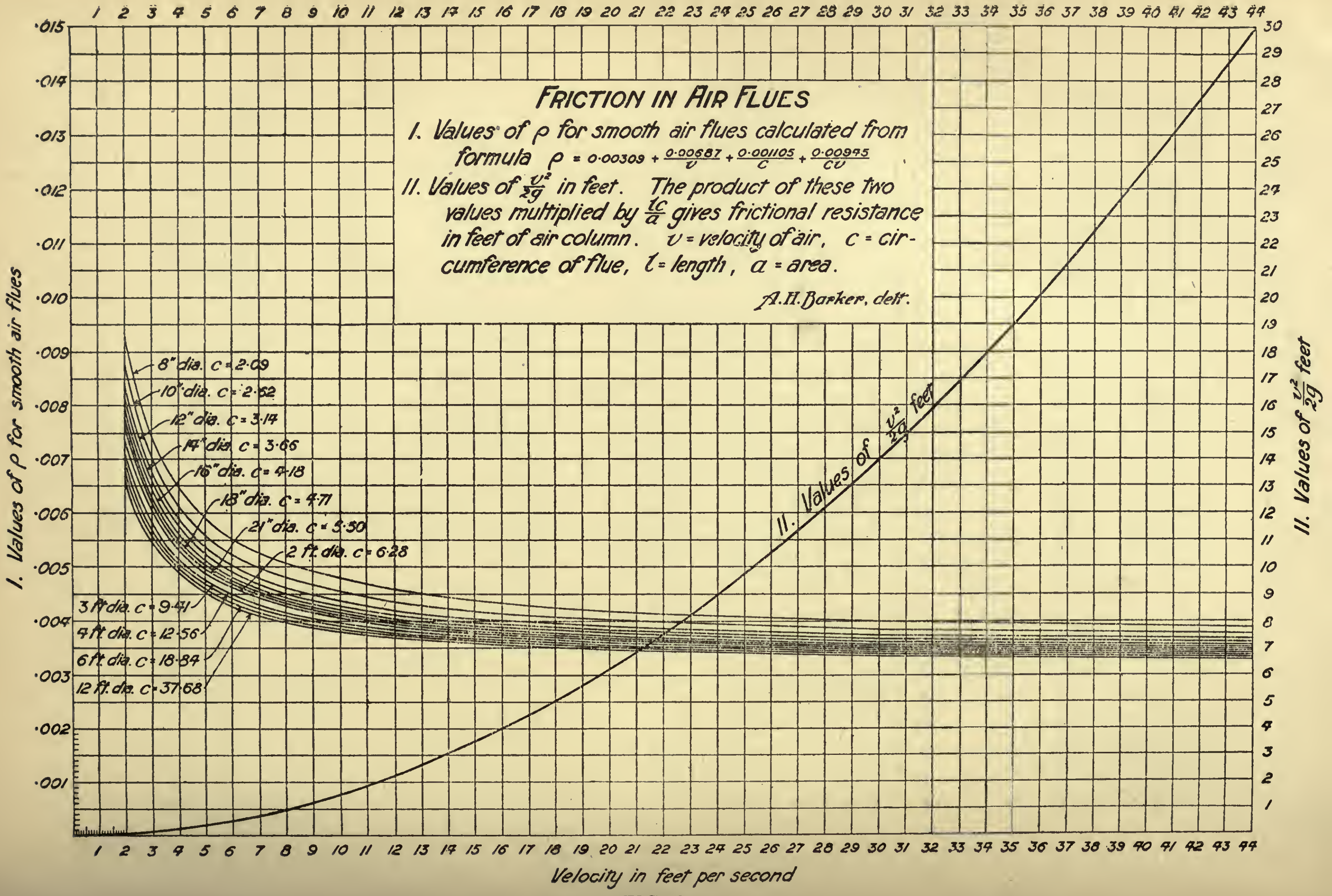

FRICTION IN AIR FLUES
I. Values of ρ for smooth air flues calculated from formula ρ = 0·00309 + 0·00687/v + 0·001105/c + 0·00945/cv
II. Values of v²/2g in feet. The product of these two values multiplied by lc/a gives frictional resistance in feet of air column. v = velocity of air, c = circumference of flue, l = length, a = area.
A.H. Barker, delt.
8" dia. c = 2·09
10" dia. c = 2·62
12" dia. c = 3·14
14" dia. c = 3·66
16" dia. c = 4·18
18" dia. c = 4·71
21" dia. c = 5·50
2 ft. dia. c = 6·28
3 ft. dia. c = 9·41
4 ft. dia. c = 12·56
6 ft. dia. c = 18·84
12 ft. dia. c = 37·68
II. Values of v²/2g feet
I. Values of ρ for smooth air flues
·001
·002
·003
·004
·005
·006
·007
·008
·009
·010
·011
·012
·013
·014
·015
II. Values of v²/2g feet
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30
Velocity in feet per second
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44

FIG. 8a.

(2) To determine the length of the sides of a rectangular flue equal in carrying capacity to a square flue of side x.

Let y and z be the sides of the rectangular flue, then with the same notation as before, we have

$$x^2 v_1 = y z v_2$$

$$v_2 = \frac{x^2}{y z} v_1$$

Also

$$\frac{4 \rho_1 l x}{x^2} \frac{v_1^{\ 2}}{2 g} = \frac{2 \rho_2 l (y + z)}{y z} \frac{v_2^{\ 2}}{2 g}$$

$$\frac{2 \rho_1}{x} = \frac{\rho_2 (y + z)}{y z} \frac{x^4}{y^2 z^2}$$

or again taking

$$\rho_1 = \rho_2$$

$$2 y^3 z^3 = x^5 (y + z).$$

From these equations and the given diagram any problems relating to required sizes of flues for any given drop in pressure, corresponding to a given delivery, can be at once solved.

In cases where the desired point is between two of the given curves on the diagrams, an estimate of the exact value of the round flue required can be made by reference to the proximity of the point to two curves. This, without substantial error, may be taken in proportion to its distance from either curve. Thus, if point is 25% (or about a quarter) of the distance between, say, curves representing 12″ and 14″, it indicates that the desired diameter of the round flue would be about 12½″, and an equivalent square flue would be 11.4″, and an equivalent rectangular flue, one of whose sides y is, say, 15″, would have the length of the other side z revealed by the solution of the equation

$$2 \times (15)^3 \times z^3 - (12.5)^5 .15 - (12.5)^5 z = 0.$$

These cubic equations would be most easily and rapidly solved by trial.

An approximate value for the second unknown side can be ascertained by inspection and corrected after substitution in the equation until a fairly correct value is obtained.

In order to obtain accuracy, the determined flue may be separately calculated from the formula, but for practical work it will generally suffice if flues are determined slightly, but very slightly larger than the calculated quantity.

Flow of Water in Pipes.

(1) Weisbach's Theory.

The heating engineer is only concerned with the flow of water in round pipes, for which the general formula is as already shown on page 88.

Loss of head in feet of water =

Starting head.	Resistance of straight part.	Resistance of single obstructions.
$\frac{v^2}{2g}$ +	$\frac{4\rho l}{d} \times \frac{v^2}{2g}$ +	$\Sigma \zeta \frac{v^2}{2g}$

The form of the law governing the flow of water in pipes is the same as for the flow of air in flues, the difference being merely in the co-efficients. It should be noticed that since in the case of water, only round pipes are in question, the value (4ρ) is used as the practical co-efficient, and is four times the value of the theoretical co-efficient ρ.

Starting Pressure.

In order to start a continuous stream of water from rest and to maintain it at a constant velocity, the amount of pressure necessary is $\frac{v^2}{2g}$ which expression gives, as before, either the pressure in feet of water column, or (what is the same thing) the amount of energy in foot-pounds per pound of water. Table IX.*a* shows the value of the expression of $\frac{v^2}{2g}$ for a graduated set of velocities, and the value is already set out graphically on the curve on Fig. No. 8*b*.

It will be seen that the actual pressure in inches of water required to start and maintain a stream of water at a moderate or low velocity in a pipe is quite insignificant compared with the amount of head generally available, and with the resistance pressure of quite a short length of pipe.

Resistances of Straight Lengths.

Value of ρ.

The values for ρ given by Weisbach, reduced to English measure, such that the result will be given in feet of water column when all dimensions are given in feet, are as follows :—

$$\rho = 0.001439 + \frac{.0172}{\sqrt{v}}$$

where v = velocity of water in feet per second.

Where, however, the velocities are given in inches per second the formula is

$$\rho = 0.01439 + \frac{.0595}{\sqrt{v''}}$$

It will be remembered that the frictional resistance in a pipe was at first stated to be proportional to the square of the velocity, and that the form of the formula is therefore given as

$$p = \rho \frac{l}{d} \times \frac{v^2}{2\,g}$$

Units.

It has been previously explained that the value of ρ varies according to the units in which the result is expressed. The natural unit in which to express the resistance to the flow of any kind of fluid through a conduit or pipe is in terms of the height of a column of the same fluid which would produce the pressure which it is required to express.

If v is expressed in inches per second, and the value of g is also expressed in inches per second per second, the resulting value of $\frac{v^2}{2\,g}$ will represent, in inches, *the same height* as when $\frac{v^2}{2\,g}$ is calculated in feet or in metres, or in any other units whatever, the numerical value of g being in all cases expressed in the same units of length as the velocity.

Now if the formula

$$p_1 - p_2 = \frac{v^2}{2\,g} + \rho \frac{l}{d} \frac{v^2}{2\,g} + \Sigma\,\zeta \frac{v^2}{2\,g}$$

is to be employed, it is evident that each term of the right hand side must be expressed in the same units, otherwise the formula would be meaningless.

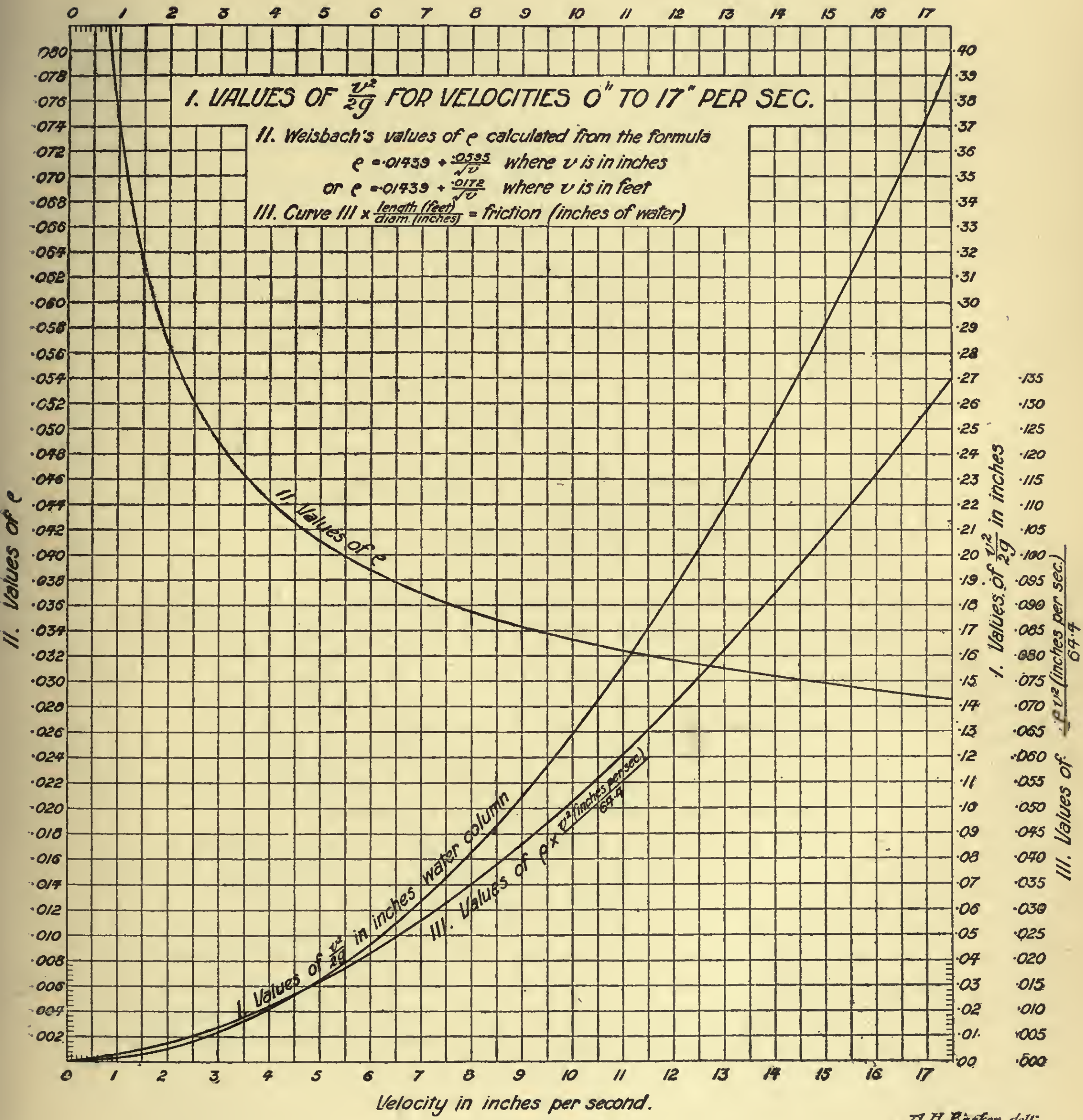
If $\frac{v^2}{2\,g}$ represents feet of fluid column, then

$$\rho \frac{l}{d} \frac{v^2}{2\,g} \text{ and } \Sigma\,\zeta \frac{v^2}{2\,g}$$

must also represent feet of fluid column.

If $\frac{v^2}{2\,g}$ is expressed in inches or metres, then the other terms must likewise be expressed in terms of inches or metres.

Now for all work in English units relating to the flow of water in pipes, the inch water column is the most convenient unit of pressure. It is very



A. H. Barker. delt.

FIG. 8b.

desirable that whatever the terms in which ρ is expressed, the results should be given in inches of water column. This involves that v should also be in inches per second, and g in inches per second per second

The value of g = 32.2 feet per second per second = 386.4 inches per second per second.

But in practice it is much more convenient to express lengths of pipe in *feet* and the value of g by the familiar 32.2 instead of the unfamiliar and cumbersome 386.4. As it happens, both these can be employed in the formula without altering its value.

The reason, of course, is that if the natural value of the factor l of the numerator is divided by 12, then in order to preserve the same value for the whole expression, the natural value of a factor g of the denominator must also be divided by 12. This is merely a device to secure convenience of calculation.

The adopted values of ρ, therefore, are such that the expression

$$\rho \frac{l}{d} \frac{v^2}{2\,g}$$

gives the friction expressed in inches of water, when

l is the length in feet.
v is the velocity in inches per second.
d is the internal diameter in inches.
g is the acceleration, due to gravity, in feet per second per second.

These values of ρ are as given in the appendix table along with the corresponding values of $\frac{v^2}{2\,g}$ and the produce of $\rho \frac{v^2}{2\,g}$. They are also set out on the curves in diagram No. 8*b*.

These values of ρ are therefore a resumé of the results of Weisbach's experiments on straight pipes, and they apply, presumably, to the pipes of the actual quality used by Weisbach in his experiments.

In order to calculate the resistance of a pipe of given length and of given diameter, when a given quantity of water flows along it, it is necessary to ascertain from the diagram, Chapter XXII., what is the corresponding mean velocity of the water along the pipe. The table or diagram will then give the corresponding values of ρ and of $v^2 \div 2\,g$ and also their product.

Take the product, multiply by the length of the pipe in feet, and divide by the diameter of the pipe in inches. The result will be the loss of pressure in inches of water column produced by the flow.

The calculation of the resistances of a pipe from these formulæ is, however, a somewhat tedious business, and in order that they may be conveniently used, Table X. and Figs. in Chapter XXV., have been calculated and plotted, showing the ratio between the quantity of water and the loss in pressure in inches of water column for all standard sizes of pipe for a 10-foot run of straight pipe. This length of pipe has been selected as the standard unit of length for this purpose, since one foot is too small a unit for convenient handling.

The result of using one foot for the unit length of pipe is that the resistance per unit length has a microscopic value, and this tends to give a sense of unreality to the results and leads to mistakes in the calculation.

The resistance for one foot can of course be easily attained by shifting the decimal point one place to the left. Two sets of diagrams have been prepared. The first two of these show the results for low velocities, such as are common in ordinary low pressure hot water heating, without forced circulation. The second pair of diagrams show corresponding results for higher velocities, such as are common in accelerated systems of water circulation.

Single Resistances, Value of ζ.

ζ is the proportion of the total kinetic energy of the water which is destroyed when the stream passes resistances of various kinds.

The values given by Rietschel are as follows :—

Kind of resistance.	Value of ζ
Sharp elbow	1
Round elbow	0.5
Return bend	0.8
Sudden enlargement	1.0
Open straightway valve (Peet pattern) ...	0
Small increase of area (*e.g.*, the enlargement in C.i. elbow)	0
Contraction	Depends on nature
Open cock	0.1 — 0.3
Open valve, ordinary seated	0.5 — 1.0

Other Formulæ.

It has been previously explained that there are a large number of other formulæ in existence, which purport to represent the values of resistance to flow of water in pipes.

One serious drawback to the use of Weisbach's formula from a practical point of view is that although the value of each term in the expression for total resistance is made to depend on the square of the value of the velocity, it makes the two last expressions depend on different powers of the velocity. For one term of the formula for the value of ρ

varies inversely as the square root of the velocity, and therefore the resistance of a single obstruction, such as an elbow, cannot be expressed as exactly proportional to the resistance of a certain length of the straight part of the pipe.

The consequence of expressing the result in this form therefore is that entirely separate calculations have to be made for the resistance of straight runs, and the resistances of single obstructions. It would obviously be much more convenient for practical purposes if the single obstructions could be expressed in terms of straight lengths of pipes, so that the existence of an elbow in a pipe would merely have the same effect as adding a certain length of the same size of pipe to the actual length, and this modification of the formula is therefore desirable in the interests of practical calculations.

It will be shown later that the degree of accuracy to be obtained from these calculations is not so great that a small theoretical inaccuracy in one item need stand in the way of so desirable a change in the method of calculation.

The consequence of this modification of the theory is, as stated above, that the resistance of obstructions can be given as a certain length of pipe, which being added to the actual length of pipe, gives a result very near to the actual total resistance. Table 13 gives the value of these lengths.

Formula of Poiseulle.

At very low velocities, such as are used in low pressure hot water heating, below $\frac{0.204}{d} \times \frac{\eta}{\gamma}$ metres per second, the resistance is expressed according to Poiseulle as follows :—

$$p = 0.0003262 \frac{l\,v}{d^2} \times \frac{\eta}{\gamma} \text{ in metres of water column.}$$

For velocities larger than this, namely, between the limits

$$\frac{0.204}{d} \times \frac{\gamma}{\eta} \text{ and } \frac{11.2}{\sqrt{d}} \times \frac{\eta'}{\gamma}$$

the loss of pressure can be expressed as follows :—

$$p = \frac{l\,v^2}{250\,d} \left(0.33 + \frac{0.036}{\sqrt{d}}\right)$$

metres of water column where

p = pressure lost in metres water column.
l length of pipe in metres.
v velocity of water in metres per second.
η = absolute co-efficient of tenacity.
γ = weight of 1 c.c. of water in grammes.
$\frac{\eta}{\gamma}$ Absolute modulus of tenacity.

The following table gives the values of η and $\frac{\eta'}{\gamma}$ and the consequent values of the expression :—

$$\frac{0.204}{d} \times \frac{\eta}{\gamma}$$

and

$$\frac{11.2}{\sqrt{d}} \times \frac{\eta}{\gamma}$$

expressed both as in velocities as metres per second, and in inches per second.

FORMULA OF POISEULLE.

Values of η and $\frac{\eta}{\gamma}$

Temp. °F.	0°.	50°.	68°.	86°.	104°.	122°.	140°.	158°.	176°.	194°.	212°.
γ	0.01775	0.01310	0.01010	0.00805	0.00660	0.00550	0.00465	0.00395	0.00350	0.00320	0.00298
$\frac{\eta}{\gamma}$	0.01775	0.01310	0.01011	0.00808	0.00665	0.00556	0.00473	0.00404	0.00361	0.00332	0.00311

Table of limiting velocity $\frac{0.204}{d} \times \frac{\eta}{\gamma}$

Calculated for a temperature of $\left\{\frac{113^\circ \text{ F.}}{45^\circ \text{ C.}}\right\}$ where $\frac{\eta}{\gamma} = .006.$

Dia. of pipe.	$\frac{3}{8}$″.	$\frac{1}{2}$″.	$\frac{3}{4}$″.	1″.	$1\frac{1}{4}$″.	$1\frac{1}{2}$″.	2″.	$2\frac{1}{2}$″.	3″.	4″.	5″.	6″.
Vel. in m. s....	0.111	0.087	0.061	0.049	0.036	0.031	0.025	0.019	0.016	0.012	0.009	0.008
„ „ ins....	4.38″	3.43″	2.41″	1.93″	1.42″	1.22″	0.985″	0.75″	0.63″	0.475″	0.355″	0.315″

Table of limiting velocities $\frac{11.2\ \eta}{\sqrt{d}\ \gamma}$

Dia. of Pipe.	$\frac{3}{8}$″.	$\frac{1}{2}$″.	$\frac{3}{4}$″.	1″.	$1\frac{1}{4}$″.	$1\frac{1}{2}$″.	2″	$2\frac{1}{2}$″.	3″.	4″.	5″.	6″.
Vel. in m. s....	0.640	0.569	0.476	0.425	0.365	0.339	0.304	0.263	0.239	0.246	0.185	0.170
„ „ ins....	25.2″	22.5″	18.8″	16.7″	14.4″	13.4″	12″	10.4″	9.4″	8.1″	7.3″	6.7″

In order to illustrate the difference which exists between different formulæ derived by different experimenters, and to show how two apparently different formula may give results which, if not approximately the same, are at least of the same general character, the results of the Weisbach formula and Poiseulle's formula have been plotted on the same diagram in Chapter XXIV., where Weisbach results are shown in firm lines and Poiseulle's in dotted lines.

The Poiseulle's formulæ are especially useful as showing the difference which is introduced into the flow, owing to the change in the viscosity of water due to differences of temperature.

Variation in Friction Owing to Difference of Temperature.

As will be seen by examining the formula of Poiseulle, experiments on the flow of water at different temperatures in the same pipe show a slight variation in the delivery, and in the friction corresponding to differences of temperature, and showing that the water becomes less viscous at a higher temperature. But the variations between different experiments on pipes of different make vary so considerably as entirely to mask any slight difference due to difference of water temperature, as observed in experiments at different temperatures in the same pipe. It therefore appears that to introduce complications into the formula in consequence of this physical law, relative to the viscosity of water at different temperatures, would be waste of time, as far as the practical application of the law is concerned.

In consequence of these considerable variations in the friction observed by different experimenters, it is necessary to take a somewhat high value for the friction co-efficient, in order that the results obtained may in all cases be on the safe side.

As far as the question concerns the heating engineer it will be shown in a later chapter that the object of the calculation is to ensure that a certain *minimum* quantity of water shall pass through each pipe in a certain time under certain conditions. It would be a serious matter if *at least* that quantity of water did not pass through the pipe. It *is* a drawback, but a minor one, if an *excess* of water passes through the pipe. In the latter case, the nature of the drawback is, firstly, that the pipe is more expensive than it need be, and, secondly, that a pipe which passes more than its calculated proportion of water is likely to have a detrimental effect on the flow through other pipes in the same system. It is no drawback so *far as the pipe itself is concerned,* but rather an advantage that it should pass as much water as possible.

A further point is that if a pipe passes too much water it is a relatively easy thing to reduce the amount of water by inserting a disc or arranging some artificial resistance in the pipe. But if a pipe passes too little water the pipe itself may have to be replaced by a larger one, or, in the alternative, a number of resistances may have to be introduced into other pipes, so that the pipe in question shall receive its due share. For these reasons it is obvious that the object of close calculations is primarily to ensure (1) that each pipe shall be capable of passing at least the amount of water needed, (2) that the water shall be proportionately distributed between the various pipes, and it is only a secondary object to ensure that a pipe shall not be capable of carrying too much.

The corollary of this proposition is that in selecting a pipe for a given duty, if any doubt exists, choose the larger of the two, also in preparing a set of tables for practical use, use co-efficients which are high, but not too high, that is, co-efficients which correspond to the highest value found by any reliable experimenter for commercial sizes of pipes.

CHAPTER VI.

HYGIENE OF VENTILATION.

GENERAL REMARKS.

Need for Ventilation.

THE need for ventilation may be said to arise from one or both of two causes, which may be, and probably are, to some extent inter-connected.

I.—It may be necessary to ventilate a room because in fact the breathing again of air, a large proportion of which has already been breathed once, exerts an unhealthy influence on the human organism, for some reason which remains to be discussed.

II.—Ventilation may also be necessary merely in order to promote the comfort of the occupants of the room.

It is quite conceivable that air which is perfectly healthy to breathe might cause extreme discomfort to people breathing it. It is equally possible that in many cases air which is not unpleasant to breathe is extremely deleterious.

On the other hand, it is quite reasonably possible that the mere fact of any particular condition of the breathing air producing discomfort is generally either an indication or a cause of those conditions being deleterious. Nature frequently, though not always, makes deleterious things unpleasant.

It may be that the persistence of unpleasant conditions lowers the tone of the entire system and renders it less able to withstand disease.

The only fact that does appear certain and clear is that from some cause or another, whatever it may be, it is more or less injurious to any animal to breathe air, a considerable part of which has been previously breathed.

There can be no doubt that loss of health and increased liability to certain diseases is associated with living in such air.

Unhealthiness of Respired Air.

It has not yet been definitely settled what is the feature of vitiated air which does in fact, or which appears to, produce results prejudicial to health, nor of fresh air which has the contrary effect. Many of the theories which have been universally accepted up to the present date have been proved untrue by recent experiments.

CO_2.

It has apparently been definitely proved by recent work that the presence of an increased quantity of CO_2 in respired air has no deleterious effect within certain very wide limits.

It has been known for many years that persons engaged all day at work in mineral water factories or in breweries, breathing air which contains a relatively large proportion of CO_2, suffer no ill effects whatever. It has certainly been long disproved that the action of CO_2 on the organism justifies it being classed as an active respiratory poison. Indeed, variations in the amount of CO_2 breathed within certain very wide limits, say, up to 300 parts per 10,000 of air, appear to make very little difference. It does not affect to the slightest degree the concentration of CO_2 in the blood. The reason for this is that the presence of CO_2 in the blood acts on one of the wonderful self-regulating mechanisms which exist in the body, excites the breathing centres, and automatically causes the breathing to be deeper and stronger, the effect of which is that the quantity of CO_2 in the blood remains absolutely constant in spite of large variations in the concentration in the air breathed.

Oxygen.

The theory that the ill effects are due to the relative diminution in the quantity of oxygen in the air has also been definitely proved to be equally untenable. A large reduction in the quantity of oxygen normally present in air, *e.g.*, from 15 to 20% is now known to have very little effect on a healthy subject.

As explained in the last chapter, normal air contains about 21% of oxygen. A candle is extinguished in air containing less than about 17%, but the percentage can be reduced to about 14 without any apparent effect on the bodily functions. In some subjects, however, such a diminution, or its equivalent in drop of atmospheric pressure, produces the train of symptoms known to climbers as "mountain sickness."

If the proportion is reduced to 12% the breathing becomes altered, with 10% the colour of the face of a person breathing it becomes of a leaden hue, the heart palpitates, and bodily and mental activity become difficult (Dr. Leonard Hill). At about 6% consciousness is lost, and death takes place when the oxygen sinks to 3 or 4%.

Expired air direct from the lungs contains about 16% of oxygen and 5% of CO_2.

It is also untrue that considerable excess of oxygen in the breathing air is dangerous to health. The theory has long been held that such an excess produces too rapid oxidation of the body tissues. But it has now been proved that the body cannot be made to burn quicker like a fire by a greater supply of oxygen. Even air composed half of oxygen and half of nitrogen has no such effect. A great excess, however, is poisonous, produces inflammation of the lungs, and arrests the metabolism of the body. It has been proved safe to breathe air composed of one half of oxygen for an hour at least under a pressure of three atmospheres. In one cubic foot of such air there would be 1½ cubic feet of pure oxygen, measured at the ordinary atmospheric pressures.

Ozone.

The theory, too, that the healthy action of sea breezes is due in any degree to the presence of ozone is also completely disproved; the actual proportion of ozone in sea air is no greater than in any other air. What is commonly taken to be ozone in sea breezes is generally the smell of decaying seaweed.

The theory that the presence of ozone in air causes the oxidation of deleterious organic matter and the destruction of micro-organisms, appears also to be completely disproved. That concentrated ozone does produce this result is undoubted, but the degree of concentration of ozone found in ordinary air is microscopic and has no such effect.

Air charged with ozone to such an extent as to have any effect on the micro-organisms contained in it, would be so strong as to produce inflammation in the lungs of any person breathing it.

Organic Impurity.

Again, there is no proof whatever of the existence of any poisonous organic impurity in the air of crowded rooms. It was long believed that some organic substance was given off by the lungs which was poisonous when absorbed in the blood. There is no proof whatever of this theory. Indeed, it may be said to be disproved unless the quantity of this presumed deleterious substance is so small that it cannot be detected by present methods of chemical analysis (Haldane).

Germs.

Air expired by healthy persons is almost entirely free from particulate matter. No doubt persons affected with various diseases may and do give off germs with their breath. When a consumptive person coughs, or even speaks, such germs may be projected to great distances, and become widely diffused in a very large room in a very short space of time. The mere act of speaking is sufficient to project such particles to a distance of 15 or 20 feet.

In all air, therefore, it is unavoidable that there should exist disease germs in large numbers. Such germs have no power to affect a healthy body, since the counteracting influences provided by Nature are so powerful. If, however, the strength of a person is reduced by living in unhealthy surroundings, by bad food, and other causes, the resisting power diminishes, and the person may become affected by the disease in consequence. Also the relative numbers of the germs in air has a great effect. The constitution of a person who would be strong enough to resist the effect of air with a certain number of germs per cubic foot might be unable to offer effective resistance if the number of germs were increased.

The immediate loss of appetite, discomfort, and headache, etc., produced by bad ventilation cannot have anything to do with infective organisms. The effect of the latter can only operate after a considerable period.

Probable Cause of Ill Effects of Bad Air.

According to the most recent speculations it is the humidity, temperature, smell, the absence of movement, and the presence of disease germs, which are among the principal causes, which, so far as is at present known, produce evil results.

Excessive humidity and temperature and stagnation of the air, operate by reducing the rate at which air normally abstracts the surplus heat from the human body. The same features and the unpleasant smell also operate prejudicially by exciting the cutaneous and olfactory nerves. Nervous health depends on the constant instreaming of sensations and the reactions produced by these are most important (Dr. Leonard Hill).

The mechanism by which the temperature of the human body is regulated is singularly perfect, provided the conditions of the air are suitable. The mechanism consists of means whereby a greater or less amount of evaporation is promoted from the surface of the skin. The body is, in fact, in the same condition as a wet bulb thermometer and is cooled by the same means.

If, therefore, the air is at too high a temperature and is stagnant, or contains too much moisture, it prevents the necessary quantity of heat from being abstracted from the body in this way, and this is thought to be among the chief causes of bad effect produced by vitiated air.

Moisture in Room Air.

As stated above, the air exhaled from the lungs of a human being is generally in a state bordering on saturation. When, therefore, this air is at all cooled, either by being breathed into surrounding air or on to any surface cooler than itself, some of the moisture is, or may be, deposited. The breath is therefore generally just visible if the air into which it is breathed is at all damp. If the surrounding air is very dry the moisture is generally immediately taken up by the air, and the vapour in the breath is not visible.

The presence of an excess of water vapour in air probably is the cause, or one cause, for the feeling of lassitude which bad air produces, while a defect in the relative humidity causes the parched or dried up feeling which is so common a complaint against central heating systems.

The presence or absence of moisture in the air has another effect, in causing or preventing a feeling of cold. The thermometer is not by any means to be taken as an infallible criterion of the comfort of a room as regards warmth. A room, the air of which contains a good deal of moisture, always feels warmer than the same room at the same temperature containing less moisture, the reason being that relative absence of moisture in the air causes more rapid transpiration or loss of heat, owing to the increased rate of vaporisation of dampness on the body surface thereby caused.

It is probably due to the greater humidity of the English climate, that whereas in England an internal temperature of 60° is sufficient to satisfy requirements, yet on the Continent and in America, where the air is normally drier, temperatures of 68° and 70° are demanded by public opinion.

The disadvantages attending on an improper degree of humidity in breathing air are said to be that if the air is too dry it takes up moisture from every object that will yield it, causes a shrinking of boards and beams in houses, produces cracks in floors and around doors, etc., damages the furniture by shrinkage. The effect on the human organism is that too dry air absorbs moisture from the skin, from the nasal passages, the mouth, the throat, and the lungs, making the skin dry and rough, and aggravating throat and catarrhal troubles, and in some cases the developing cause of such disorders.

The effect is made worse by the fact that persons are as a rule constantly passing into and out of rooms in a very dry condition to the outside air, which may possibly be very moist, owing to its lower temperature. This is probably the cause of cold and inflammation of the throat and bronchial passages.

The failure of the vocal organs of public speakers and singers is probably the result of breathing very dry air. Dry air produces in some persons a feeling of lassitude. It has a tendency to lower the temperature of the skin too rapidly, owing to rapid absorption of moisture, and produces a feeling of coldness even in warm air.

On the other hand, air that is too damp interferes with the normal loss of heat from the bodies of persons, and again causes a feeling of oppression.

The importance of humidity as a factor in the hygiene of ventilation is emphasised in the Report of the Home Office Royal Commission on Humidity and Ventilation in Cotton Weaving Sheds.

The weaving of certain kinds of cotton cannot be successfully accomplished except in a very moist atmosphere, and the complaints of the operatives relative to the discomfort caused by working in such an atmosphere led to the appointment of a highly expert technical committee to investigate the matter. This committee having taken evidence from some of the leading scientists issued a most illuminating and valuable report.

A ballot having been taken of the operatives engaged in this industry proved almost unanimous in favour of the abolition of excessive humidity, as being, in the opinion of the operatives themselves, detrimental to comfort and health. Investigations into the matter, undertaken by the Departmental Committee, led to the conclusion that there was no evidence to prove direct injury to health as a consequence of excessive humidity, but the bodily discomfort was unquestionable, and the committee expressed the opinion that a long continuance of this bodily discomfort would probably result in injury to health.

The result of the investigation into the actual humidity in the majority of weaving sheds, of which complaint had been made, showed that it varied between 70% and 80% of saturation. The injury to health of the operatives, if any, was partly due to the chilling effect produced when leaving the hot, damp atmosphere of the shed. It was believed that this was due to the perspiration and absence of evaporation, causing dampness in the clothing worn, rather than the deposit of moisture in the clothing taken off.

The general conclusion arrived at was that the degree of discomfort experienced by workers in a hot, moist atmosphere is measured not by the temperature of the air, nor by its *relative* saturation, nor the absolute percentage of aqueous vapour present, but by the *temperature shown by the wet bulb thermometer*. If this exceeds about 78° F., hard work becomes impossible. Above about 88° F. it becomes impossible for ordinary persons even to stay in still air. If the air is in motion the limit extends upwards by several degrees.

The reasonableness of making the reading of the wet bulb thermometer the criterion will be seen from the fact that the body itself is in the same condition as a wet bulb thermometer.

It is accepted by the great majority of physiologists that a temperature of 75° wet bulb should not be exceeded, and that a limit of 70° is still more desirable, and below this temperature there should always be a minimum difference of 2° between the wet and dry bulb thermometers. The diagram will show the real physical meaning of this condition.

The effect of working in a warm moist atmosphere is to increase the body temperature, pulse, and loss of moisture, out of all proportion to the work done. Efficient work cannot be performed unless the temperature of the body is prevented from rising above a certain maximum.

If the wet bulb reading rises beyond 88° in fairly still air, the body temperature cannot be prevented from rising seriously, even in persons stripped to the waist and doing no work.

If the air is very dry a temperature of 75° to 84° can be easily endured, while hard work is being done.

Medical evidence also goes to prove that a dry atmosphere with a large carbonic impurity is preferable to a humid atmosphere with a relatively low carbonic impurity.

Dr. Leonard Hill believes that the ill effects of crowded rooms are due to the heat and humidity and nervous fatigue from excessive illumination and excitement, that it is very doubtful if the unpleasant smelling exhalations from the bodies of men have any effect on persons accustomed to them and not of æsthetic temperament, and that moving air has a very stimulating and healthful effect on metabolism.

Dr. Boycott states that the essential difference between a fresh and a stuffy room is that in the former the air is not all at the same temperature, but is broken up by draughts, which impinge on the hands and face, lower the skin temperature, and so produce a feeling of liveliness. That cool air, free from unpleasant smells, but containing 300 parts per 10,000 of CO_2, is distinctly invigorating.

To bring about relief in heated rooms by cold ventilation is dangerous, and should be avoided while the rooms are occupied by persons who have been overheated by an excessive temperature. Overheating must be avoided at all costs. This can generally be effected by periodical ventilation of the rooms during untenanted hours.

The unpleasant smells present in dwelling rooms result chiefly from the decomposition of matter from the skin, mucous membrane, and also on the clothes of the persons present. No injurious effect to health has

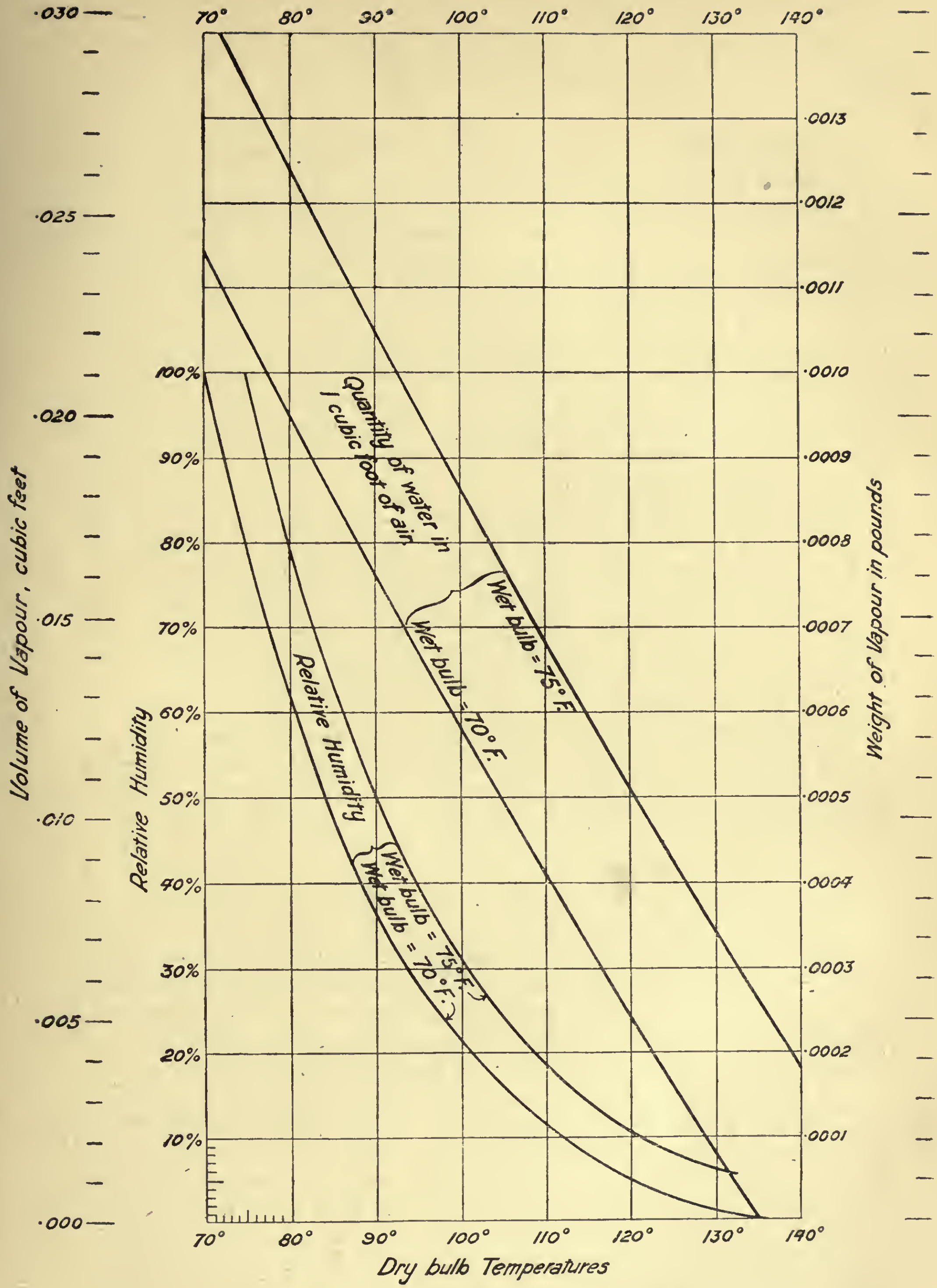


Diagram illustrating condition of air as to humidity when wet bulb thermometer is at 70° and 75° with varying dry bulb temperatures

A. H. Barker, delt.

FIG. 8c.

been demonstrated, but the smells cause a feeling of nausea on entering the room, which may be detrimental.

The results of all these investigations must not be considered as in any way in favour of a decreased rather than an increased ventilation of ordinary living rooms, lecture halls, and public buildings. That it is a great advantage to everybody to breathe plenty of pure fresh air cannot be denied. The germicidal properties of atmospheric air and sunlight are of the utmost importance in preserving our health.

The chemical composition of the air is only of importance when the impurities are large enough to make the quantity of oxygen insufficient for daily needs. Impurities can never accumulate to a really serious degree in any ordinary room in which proper attention is given to the moisture and the temperature.

It is believed that in cases where the CO_2 is derived from combustion, a standard of 20 volumes of CO_2 per 10,000 is no more detrimental than is a CO_2 content of 9 parts in 10,000 derived from human breath.

The water used for humidifying air should be of great purity. The method of humidifying by injecting steam into the air is undesirable, inasmuch as it not only raises the temperature but is apt to introduce into the air organic impurities distilled from the water introduced as feed water into the boiler supplying the steam.

In view of the increasingly important place assigned by physiologists to the humidity of the atmosphere, several diagrams have been prepared by the Author, setting out the characteristic curves of atmospheric humidity in several different aspects.

Dust.

One of the evil effects of breathing vitiated air, or indeed the air in any room between which and the outside air there is no consequent interchange, is that all interior air is more or less polluted with dust. This dust is partly inorganic, derived from the wearing away of objects of the room, and partly carried into the room by air currents from the street.

It also partly arises from the clothes and bodies of persons who have been in the room. It thus contains a certain proportion of infective bacteria. There can be no doubt that the breathing of air which is very dusty predisposes to diseases of the lungs, and indeed may directly cause such diseases, especially in cases where the dust is of a poisonous or irritating nature.

It is, further, abundantly obvious to any person who examines the path of a beam of sunlight passing through the room that there are vast numbers of microscopic dust particles floating about in the air of a room, and that this number is immensely in excess of the corresponding number floating about in the open air.

Carnelly, Haldane, Anderson, give the following as the number of micro-organisms per litre of air :—

Outside air, 0.8 micro-organisms per litre.

Elementary school, with natural ventilation, clean, 91; average, 125; dirty, 198.

Old elementary schools, opened over 20 years, 311.

Dwellings in Dundee, clean, 10; average, 34; dirtiest, 93.

The presence of a large number of bacteria in the air may be important or not, according to the nature of the bacteria. Very large numbers may exist in factories, and be in themselves comparatively harmless. The largest number of germs found in a factory by the above-mentioned authorities was 805 per litre, in a rope factory. There was no reason to believe that these germs were harmful except as dust.

The average taken over a large number of

Work-rooms was 10 micro-organisms per litre.
Elementary schools in Dundee, 152.
Country Board schools in Scotland, 76.
Poor working class dwellings, 45-60.
Better class dwellings, 9.

These figures show the close connection between the cleanliness of a room and the purity of the air.

The most dangerous form of dust is that arising from the expectoration of sufferers from phthisis, which, if allowed to dry and float about in the air, is of the most dangerous possible character.

The presence of dust or any form of dirt in the outside air has an indirect effect, in that it leads to the closing of windows and to the general restriction of the air supply. The trouble caused by the presence of smuts and black particles in the air is so great that the great majority of ordinary people would prefer to have less air rather than be subjected to the annoyance of dust.

It is obvious, too, that any system of ventilation which promotes the removal of dust floating about in the air is to be preferred to one which merely renews the air and allows the dust to remain.

Special Sources of Impurity.

These arise from many sources and cannot be generally dealt with. Many manufacturing processes produce fumes and evil smelling vapours.

The air, too, may be contaminated from sources outside the room, such as badly kept urinals or water closets. Smells thus arising, though not in themselves actively poisonous, do certainly operate prejudicially on the general health of persons exposed to them.

Other sources of contamination are escaping gas, which is especially dangerous, as it contains 7% of carbonic oxide. Even such a small percentage of the latter gas as 0.3 parts per 10,000 would produce headache, and 20 parts is dangerous to life.

Effect of Change of Conditions.

It appears that comfort as well as health is promoted by constant change in the condition of the air. These changes automatically produce reactions in the working of the body mechanism, and a sense of freshness and well being, which is noticeable by its absence in the case of dull stagnant air heavily charged with moisture. Almost any cause which would produce such a change tends to create a feeling of freshness. It is probably the existence of the change and its reaction on the body rather than the character of the change which produces the result (Dr. Leonard Hill).

For instance, very bad air may be charged with odoriferous matter which is extremely unpleasant to the senses and may produce a feeling of intense discomfort. If into air of such a kind a quantity of ozone is introduced, the mere alteration of the smell, aided, perhaps, by the imagination, is sufficient to produce a feeling of freshness, apart altogether from the question whether ozone has or has not any such chemical action on the air, as is sometimes claimed for it.

The most recent speculations (Dr. Leonard Hill) as to the cause of the success of the open-air treatment of consumption, are that living in the open air subjects the patient to constant changes of temperature, heat abstraction, humidity, and other similar things. These constant changes produce an ever changing reaction on the body mechanism and excite all the organs to excessive activity, which not only tends in itself to throw off the disease, but also enables the patient to absorb large quantities of strengthening food. It may be, and is indeed quite probable, that the healthiness of an open-air life is due very largely to these constant changes of conditions which react in this manner.

Nature subjects the body of an animal, probably not without good reason, to innumerable and unceasing changes, and it is possible or even probable that this is an essential condition of bodily well-being. The engineer, on the other hand, strives to avoid change even of the smallest kind. He endeavours to keep the internal thermometer reading constant. A constant thermometer reading in nature is almost unknown. He endeavours, in deference to public opinion, to keep the rooms ventilated free from draught or appreciable air currents. Such a condition in Nature is almost entirely unknown, and it is all but absolutely proved to be one of the most undesirable conditions of breathing air.

Dr. Hill's Experiments.

Recent experiments made by Dr. Leonard Hill at the London Hospital, confirming others of Dr. John Haldane, appear to throw doubt on the question whether the estimation of the CO_2 impurity is not a wrong basis on which to estimate the perfection of the ventilation of a room.

The experiments, which are not complete at the time of writing, confirm that not only the presence of CO_2 in air, but even of organic impurities, if any exist, other than actual disease germs, are not in the least deleterious.

The experiments consisted of observing the results on the organism of human beings, consequent on sealing them up for a considerable period in an absolutely air-tight chamber of small dimensions, and allowing the air sealed up in the chamber to be rebreathed until the amount of carbonic acid and water vapour became very high, and introducing into the chamber in addition large quantities of CO_2.

In one experiment eight adults were confined in a chamber of from 200 to 300 cubic feet capacity, and the carbonic acid was allowed to reach the proportion of 500 parts per 10,000 volumes of air. Extreme lassitude was produced in the observing subjects. A powerful motor driven fan in the chamber was then put into operation, thus producing a vigorous circulation of the vitiated air. The effect was that the feeling of lassitude was immediately dissipated.

Observations were also taken on the subjects of the experiment by noting the action of the heart and other matters.

The general conclusion arrived at was that the feeling of lassitude and oppression, hitherto assumed to be produced by impurity of the air breathed, absence of oxygen, and presence of organic impurities, was in reality due to the failure of the air to abstract the surplus heat of the body with sufficient rapidity. On starting the fan evaporation of moisture from the body was promoted and the surplus body heat abstracted.

The provisional conclusions formed from the results of these experiments are that the chief condition which breathable air should fulfil is that the wet bulb temperature should not exceed 75°.

Reference to the diagram on Fig. 8*c* shows that if this condition is fulfilled when the relative humidity of the air has values as shown by the first column, the dry bulb temperature should not exceed that shown in the second column.

If these provisional conclusions are established, they involve the complete recasting of the hitherto accepted principles of ventilation, chief regard being had to the humidity and temperature of the air breathed.

Dry Bulb Temperature Corresponding to a Wet Bulb Reading of 75°

Percentage of saturation.	Dry bulb reading.	Wet bulb reading.	Absolute moisture lb. per 1,000 cub. ft.
100%	75	75	1.35
90%	77.5	75	1.3
80%	80	75	1.25
70%	83	75	1.2
60%	86	75	1.13
50%	90	75	1.06

How far these facts have a bearing on the condition which the engineer ought to strive to produce, it is not for the engineer himself to decide.

The argument from nature can be easily overdone. The life of an ordinary human being cannot be called in any pertinent sense a "natural life," and natural conditions can only be assumed to be suitable for a natural life. At any rate, knowledge of these matters is not advanced far enough to enable the engineer to substitute new criteria of entirely satisfactory ventilation for those previously adopted. He can only assume that for some reason it is necessary for the well being of the occupants of a room that a certain quantity of fresh air should be passed through the room per occupant per hour, and he has at present to accept as a somewhat crude rule of thumb the previously recognised standards and work to them until further definite knowledge is available.

Amount of Vitiation of Air by Combustion and Respiration.

The air in a closed apartment is vitiated when gas, petroleum, or other combustible oils are used for lighting or heating, if the products of combustion are allowed to mix with the air of the apartment.

It will be first necessary to note the amount of heat and of the various impurities which the presence of living beings and burning lights introduce into the breathing air. These quantities are shown in the accompanying table. These figures are taken from various published authorities on hygiene, notably Pettenkoffer. The published figures, however, vary very considerably among themselves, according to the experimenter, and high values have in each case been taken, which in case they are too high, will in all cases have their error on the safe side.

The table in the appendix gives figures from which the degree of vitiation of the air may be estimated.

The average amount of heat given off by an adult human being awake and in good health is about 400 B.T.U. per hour, of which about 120 B.T.U. is given off by contact of air, 172 B.T.U. by radiation, and the balance 108 B.T.U. in the breath and otherwise.

A child gives off about half this amount.

Breathing.

An adult man in good health takes about sixteen complete in-and-out respirations per minute. The volume of each inspiration is about from 25 to 40 cubic inches, according to the size of the man and the condition of his lungs. The inspiration, of course, draws the air into the interior of the lungs, which is provided with a multitude of air cells. There are said to be in the human lungs between five and six million of such cells, and their superficial area is said to be about 1,000 square feet.

The composition of the air when it is drawn in is about 21 parts by volume of oxygen, 79 parts by volume of nitrogen, and about 4 parts in 10,000 of carbon-di-oxide. Expired air contains, say, about 16 parts of oxygen, 79 parts of nitrogen, and 5 parts of CO_2. Healthy expired air is almost entirely free from particulate matter. Loud speaking or coughing, however, causes the projection to considerable distance of any germs there may be in the mouth or lungs.

The temperature at which this air is expired varies with the outside temperature. When the surrounding temperature is 60° to 65° the air leaves the lungs at about 95° to about 96° F. When the external temperature is 32° the air leaves the lungs at about 87° F. The expired air is in all cases practically saturated with moisture.

From the figures given in Table V. it will be seen that the amount of water vapour in 40 cubic inches of saturated air at 96° is about .0000565 lbs.

Multiplying these figures by the time, it will be seen that the approximate quantity of heat and various impurities normally given off from the body and lungs during 24 hours are as follows:—

CO_2, 22 cubic feet, or an average of 0.9 cubic feet per hour, asleep and awake.

Water vapour, 1.3 to 3.15 lbs.

Heat, 10,000 B.T.U.

Lighting.

The carbon-di-oxide from lights and stoves being liberated at a high temperature, unless special arrangements are made for withdrawing it immediately from the room air, at once finds its way direct to the ceiling of the apartment, over which it becomes distributed as a warm layer of vitiated air. As it gradually cools down it is drawn down into the breathing plane and, of course, warms the room at the same time as it vitiates it.

The proportion of the energy in gas which is converted into actual light is small and need not be further considered. The greater part of the energy is converted directly into heat. Ultimately, of course, the absorption of the light by the walls of the room converts the light energy also into heat, so that the whole energy of the gas burnt has to be dealt with by the ventilating engineer. An ordinary gas jet produces as much CO_2 as three men, and as much heat as five men.

The principal evil effect due to burning gas in breathing air is not the production of CO_2 nor water vapour, but the microscopic proportion of sulphur in the form of the disulphide which is left in almost all gas in spite of the careful efforts to abstract it which are made at the gas works. This sulphur burns to sulphurous and sulphuric acid. These are the cause of the pungent character of air vitiated by burnt gas. The amount of sulphur left in the gas varies in different towns. In London it is limited to 120 grains per thousand cubic feet, but in other towns the amount is much higher.

Composition of Healthy Air.

For the purposes of the ventilating engineer atmospheric air may be taken to contain on the average 2.5 to 4 parts of CO_2 per 10,000 volumes. The lowest percentage of CO_2 ever recorded at sea level by a reliable experimenter in atmospheric air was about 2 parts per 10,000.

This analysis was of air taken in mid ocean in a tropical climate during the rainy season. The effect of a constant deluge of rain and a wide expanse of water in dissolving the CO_2 normally contained in air, probably accounts for this low proportion. The highest percentage ever recorded in the open air, in atmosphere free from fog, excluding special cases (e.g., where a space open to the sky cannot on account of its surroundings be said to be open air in the real sense) is about 5.5 parts per 10,000.

The average air of large towns in winter contains about 4.5 cubic feet of CO_2 per 10,000 cubic feet. The maximum in badly over-crowded districts may be taken as 5 parts per 10,000.

It is probable that the average London air, when there is no fog, contains as follows :—

$$\left.\begin{array}{lr} \text{April to September} \quad \ldots \quad \ldots & 3.3 \\ \text{October to March} \quad \ldots \quad \ldots & 3.7 \end{array}\right\} \begin{array}{c} \text{Parts of } CO_2 \text{ per} \\ 10{,}000 \text{ of air} \end{array}$$

Variations of CO_2 in Atmospheric Air.

The amount of CO_2 in the atmospheric air varies considerably, according to the conditions. In summer weather the proportion may rise to 3.5 volumes by night, or fall to 2.6 volumes by day in the lower strata of the air in consequence of the influence of vegetation.

The classical estimations of the CO_2 in air made by Pettenkoffer appear to have been about 0.5 parts too high.

Determinations by Haldane of the purest country air in Scotland show an average of about 3.0 parts per 10,000.

Fog.

The presence of a fog is almost always accompanied by a very definite rise in the percentage of carbonic acid in the air, and probably also a fall in the percentage of oxygen.

Dr. Russell, late of St. Bartholomew's Hospital, on one occasion found 14 volumes per 10,000 of carbonic acid in the outside air during a very dense and prolonged fog. The usual values during an ordinary London fog are

Slight white fog, 5 parts.
Heavy white fog, 7 parts.
Dense black fog, 9 parts.
Maximum recorded, 14.1 parts.

It is evident, therefore, that no test of ventilation by means of carbonic acid estimation is of any value during a fog.

Owing, doubtless, to the same cause, the average proportion of CO_2 is about 0.4 parts per 10,000 greater in winter than in summer.

In a town the percentage is generally 0.5 parts higher than in the country.

Practical Standards of Ventilation.

It is a rough general rule that if the proportion of carbon-di-oxide in the air in a room exceeds one volume in one thousand of air, that is 10 parts in 10,000 of air, it being assumed that this carbon-di-oxide proceeds from the lungs of human beings, the air is unhealthy to breathe.

There is a great deal of difference of opinion in regard to this, and it is impossible to obtain anything like precise figures.

The following general observations contain all the information on this subject that is likely to be useful to the ventilating engineer.

As we are so uncertain as to what constitutes really satisfactory ventilation from a hygienic point of view, the ventilating engineer has to consider the standards which will be accepted as satisfactory by his clients.

It may be stated generally that if the air of a room is kept free from organic smells, the ventilation will be regarded as satisfactory, and however unsatisfactory a criterion this may be from a scientific point of view, the ventilating engineer has to accept it to some extent as a guide. But the amount of smell generated by a number of people in a room depends on the physique of the persons and on their cleanliness. There are perfectly cleanly and otherwise healthy persons, who cannot occupy a room for more than a few minutes without causing it to smell. There are likewise uncleanly persons whose bodies do not give off smells. Generally the clothes of uncleanly persons are more responsible for smells than are their bodies, and it is probable that smells, however disagreeable, while not positively unhealthy in themselves, except in so far as they may be accompanied by disease germs, by their action on the olfactory nerves produce unfavourable results on the general health.

Instances are known to every ventilating engineer of large rooms, through which enormous volumes of air are being propelled, yet have a distinctly unpleasant smell when occupied by dirty persons. The ventilating engineer cannot be sure at all times of keeping a closed space free from odour, howsoever well it be ventilated.

In an ordinary assembly of cleanly persons the sense of smell is sufficient to detect impurity in the air when the carbonic impurity is above 8 or 9 parts per 10,000 cubic feet. The same degree of carbonic impurity would, if the assembly were otherwise than cleanly, be distinctly disagreeable. If the carbonic impurity reaches 15 parts per 10,000 it feels as bad as it is possible to be, that is to say, that if the air contained 20 parts per 10,000 it would smell no worse than if it contained 15.

Instances are however known, and are not infrequent in our older public buildings, in which a more or less crowded audience produces vitiation of the air to the extent of 35 parts of CO_2 per 10,000 air, or even more. Up to 50 parts have been recorded.

In brief, therefore, it may be stated that ordinary ventilation of a crowded building should not allow a greater carbonic impurity than 10 parts in 10,000, or one part per thousand. Good ventilation may be taken to be 8 parts per 10,000; very good, 7 parts per 10,000; extremely good, 6 parts per 10,000.

The calculation of the quantities of air necessary for maintaining these standards will be dealt with in a separate chapter.

It is not possible to say how many cubic feet of air are required per head, unless the precise conditions which are to be attained are laid down, then the matter is capable of exact calculation, but the practice does not always bear out the calculation, probably because the effect of crowding on the emissions from human beings had not been sufficiently investigated.

The results to be aimed at in all systems of heating and ventilation are not only that the room shall be in fact healthy to inhabit, but also that the occupants, as far as possible, in the room shall be unconscious that any heating system exists or is absent, and equally unconscious that an adequate quantity of ventilating air is, or is not, passing. It is clear that in a matter of this kind the feelings and sensations of individuals are of the greatest importance, and it is equally clear that no uniformity of result when gauged by such a standard as individual feeling can possibly be obtained. One individual will find a room insufferably hot, when another will find the same room and at the same time intolerably cold.

The demands for a high temperature are apt to be progressive, a person accustomed to live in a temperature of 60° gradually comes to desire 65°, and one accustomed to 65° wishes for 70°.

As regards ventilation, one individual will feel that a room filled with pure warm air is intolerably stuffy, while another individual will find the same room all that could be desired. From the heating engineer's point of view, the trouble is chiefly this, that the dissatisfied person will talk and complain, whereas the satisfied person will say nothing. Under these circumstances it appears impossible that there should ever be a hall or room maintained in such a condition that nobody will complain, and if the satisfactory character of the heating and ventilation be gauged by the existence of complaints, it is fairly certain that a wholly satisfactory system is impossible of attainment. The most, therefore, that a heating engineer can hope to attain is that a definite specification shall be obtained, either from the medical authorities, from the architect, or from some other person responsible for the satisfactory character of the result obtained, and that the heating engineer should be entitled to claim that his result is satisfactory, provided that specification is met. Such a specification can in the nature of things only consist of 4 or 5 parts at most.

(1) That the temperature, as indicated by the thermometer, shall be maintained between certain limits, and with a certain degree of uniformity over the building.

(2) That a certain minimum number of cubic feet of air, as measured by the anemometer, shall be delivered to the room per hour, or that so much air shall be uniformly distributed per hour in such a way that if the room is occupied by a certain number of human beings the percentage of carbon-di-oxide shall not exceed a certain maximum at any part of the room at any time.

(3) That the quality of the air delivered shall, if the apparatus be properly attended to, fulfil certain conditions as to cleanliness and humidity.

(4) That there shall be no air in the room moving at a greater velocity than a certain maximum.

If the heating engineer can fulfil simultaneously all these conditions, he is entitled to claim that all reasonable demands on him have been absolutely satisfied, irrespective of the feelings of the occupants.

It is the obvious duty of the medical authorities, if the specification is satisfied, to ascertain what grounds, if any, there are for complaints, and if necessary to change the specification in such a way that all reasonable ground for complaint may be removed. He is then entitled to expect the heating engineer to work up to his new specification if possible, but it is certainly not the responsibility of the heating engineer to investigate these subjective conditions. The heating engineer is an engineer, not a medical man or hygienist, and the fact cannot be too strongly insisted upon that what is wanted at the present time is not so much the power of the engineer to fulfil any specified conditions as the power of the medical authorities to specify accurately what are the desirable conditions that the engineer should fulfil.

The criteria to which the heating engineer must work are the thermometer, the anemometer, the hygrometer, the carbon-di-oxide measuring appliances, and other scientific instruments, and not the self-contradictory feelings or complaints of the heterogeneous occupants of a building.

So far, therefore, as the heating engineer is concerned, he must see to it that his appliances do actually discharge and uniformly distribute in the room at a sufficiently low velocity a certain number of cubic feet of air per hour, duly warmed to the required degree, cleansed from possible impurities, and humidified to the requisite degree. The conditions, therefore, which are required to bring about this state of things are what are under consideration in the present volume.

Draughts and their Uses.

The condition of the air in a room is, of course, in ordinary circumstances chiefly controlled by the introduction of fresh air. In cold weather if fresh air is introduced unwarmed the natural result is that the temperature of the apartment is lowered too much for the comfort and well-being of the occupants. For comfort it is a prime necessity that the air should be by some means warmed before introduction. If air is introduced at any point in a room in the neighbourhood of human beings at a considerably lower temperature than the temperature of the room, it will at once fall to the floor, owing to its great specific gravity, and any person who is within a short distance of the inlet will feel the introduction of fresh air as a cold "draught." Modern civilisation has made people too susceptible to draughts. As a matter of fact a draught is in itself a very healthy and desirable thing. Persons' constitutions are however so much weakened by staying in hot moist and germ-filled air, that a health-giving draught is apt to produce chills. The true remedy, of course, is to inure the public to the gospel of fresh air.

In the present state of public education the introduction of air into a room in such a way as to enable any of the occupants to feel it, always gives rise to loud complaints of discomfort, and as this is always laid to the charge (rightly or wrongly) of the ventilating arrangements, it behoves the heating engineer, whatever his own private opinion may be as regards draughts, to suppress them at all costs. As a matter of fact draughts are generally suppressed by suppressing or diminishing ventilation. The heating engineer is the servant and not the mentor of the public, and is bound to fall in largely with the public's ideas of what is good for them, and so his chief aim in ventilating any apartment in which a number of people are to be assembled together must be as much to suppress draughts as to ventilate the apartment.

Draughts are, however, not only caused by the introduction of cold air, but also by the cooling of air already in the room by cold windows or walls, which also causes a down draught of the cooled air. This cool air is simply a portion of the air already in the apartment which has been made heavier by the increase in specific gravity, due to the cooling, and therefore falls to the floor.

Whenever an apartment is ventilated, therefore, it is of the first importance to consider carefully the direction of flow of *all* the air in the apartment, including the condensation currents and the currents generated by the introduction of the air.

If, therefore, air is introduced at a higher temperature than that of the room the current will at once rise, and this rising of the current of fresh air will cause the existing air in the apartment to follow it and circulation will thereby be caused. The warm air will rise to the roof of the apartment, where it will be gradually cooled, and will gradually find its way down by diffusion and displacement to the breathing level. Unfortunately, however, the warm air ejected from the lungs of the inhabitants also tends to find its way to the top of the building, owing to the relatively high temperature, where it mixes with the incoming fresh air, so that the composition of the air which finds its way down ultimately to the breathing level is merely a mixture of good and bad air, which has a smaller proportion of CO_2 in its composition than the remainder of the air at floor level.

This admixture, though very undesirable, is, of course, to some extent inevitable. The only way to enable the occupants of any room to breathe absolutely pure air would be to breathe it through a tube direct from the outside. This is of course entirely impossible.

Removal of Dust in Air.

There is, however, another fact of ventilation which has to be taken into account, and this, in the opinion of many prominent medical authorities, is of as much, or more, importance than the question of the chemical purity of the air. This question is that of dust, containing, as dust always does in an occupied room, micro-organisms of various kinds. If the moving of the air of the room is so slow as to be imperceptible as a draught, these micro-organisms, which, though heavier than the air, yet on account of their extremely minute size and weight, float about in the air and gradually settle on all horizontal surfaces as dust. A microscopic examination of all dust will show the presence of these micro-organisms.

Now if the room is ventilated freely without much attention to draughts, the currents of air will carry with them a large proportion of these micro-organisms, which will be ejected with the outflowing streams of air. This is a very desirable state of things, but it cannot be carried out in general practice, because of the strong objection to perceptible draughts, which the public have been taught to cultivate. A draught which is so slight as to be imperceptible has not sufficient power to remove any but the finer and lighter dust particles. The remainder, including the heavier and probably the most dangerous portion of the dust, will settle down in the room in spite of an air current which is so slight as to be imperceptible.

The method adopted for the removal of these micro-organisms is the use of the sweeping brush and duster, which devices certainly do not remove a third of the dust which they distribute. The general effect of the use of the sweeping brush is merely to set the particles of dust and micro-organisms into circulation again, and when the person wielding the broom has completed her task they simply settle down again where they were before. The only correct way to sweep and dust a room is to have a " blow through " by open windows and otherwise at the time the sweeping operations are proceeding. If this is not done the hygienic effect of the sweeping is always largely lost.

Recent introductions of suction devices for sucking the dust away are naturally a great improvement, but it may be doubted where these can effectively replace the method of circulating the dust by means of a broom and blowing it out by a draught.

The Senses as a Criterion of Ventilation.

In so far as success of a scheme of ventilation depends on the feelings of the persons occupying the room, there can be no doubt that the movement of air within a room is one of the causes which produce a feeling of freshness. If the velocity of motion of the air, at whatever temperature, is too great, it begins to be felt as a " draught " and complaint is accordingly made. If, on the other hand, the air being impure is too stagnant, at whatever temperature and from whatever cause, it is liable to cause complaint of stuffiness and oppression. It will, therefore, be clear that the movement of air for the best effect, as far as the sensations produced are concerned, must be within certain limits, and these limits may be placed at between one foot per second and three feet per second.

It is not at all necessary for the production of this feeling of freshness that the moving air shall be fresher than the general air of the room, or indeed that it shall be fresh at all. If highly vitiated air, provided it is not too damp or moist, is set in motion by a fan working in the room itself, a feeling of freshness is produced which would give place to the contrary feeling if the fan were stopped.

It has been the experience of the present writer that the feeling of freshness has little to do with the actual carbonic impurity of the air, nor with the presence of dust, but only with the degree of dampness of the air and the amount of motion. On one occasion the air of a room at 60°, which felt unbearably stuffy, was warmed up to 65° or 67° and at the same time was set in motion by means of a desk type propeller fan without any fresh air being introduced at all, and the feeling of oppression was removed. The cause probably being that the increased temperature of the apartment dried the air, that is to say, removed it further from the dew point, and the moving of air assisted the emission of heat from the bodies of the persons in the room.

Generally speaking, if the air of any room or hall is maintained at a temperature of from 60° F. to 65° F., the humidity being between 50% and 70%, and if a gentle movement of the air of the room is maintained at a velocity between one and three feet per second, and if the carbonic impurity does not exceed 10 per 10,000, the occupants of the room being cleanly persons, no complaint will be made by any of its occupants, except such persons (of whom there are always a few to be found) who would complain in any circumstances.

Cleaning of Air.

The air of a large town, or indeed of any town, is always in need of some preliminary cleansing before it is passed into a building. It contains particles of dust, soot, insects, and other impurities, which, so far as possible, should be removed from it.

In all installations in which fresh air is taken from the one place, regard should be had to the protection of the inlet against dust, foul air, smoke, soot, and other impurities, and also where the circulating power is low, against wind.

The means known to the modern science of ventilation by which this can be done consists of dry screening and wet screening of various degrees of perfection. Dry screening may be employed for removing the coarser impurities, such as minute insects, smuts, and to some extent dust. In order to remove the smaller impurities it is necessary that all these impurities shall be thoroughly wetted. This is the object sought in the various methods of wet screening. The air is passed through a screen consisting of some woven fabric, or of closely packed coke or other material which is kept continually wet by a continuous stream of water, through which the air can find its way, being constantly changed in direction in so doing.

Dry Screens.

These consist of woolly materials, not too closely woven, the total area of the space between the various strands being sufficient to allow the requisite quantity of air to pass without requiring too great a difference of pressure at the two sides of the screen.

Yet another and more perfect method of separating of particles from the air is to pass the air between layers of cotton wool placed between gauze of fine mesh, or through layers of finely powdered coke. The latter process is said to have the effect not only of mechanically separating the particles of dust, micro-organisms, etc., but also of oxidising such particles by the oxygen which the coke occludes. In all cases the total area of the filter must be as large as it can possibly be made. The slower the air passes through the filter the better will be the effect in all cases, also all such filters must be easily removable for cleaning and washing, and it is the necessity of this process which makes the maintenance of a filter of any kind somewhat expensive and a tiresome business.

The pressure which is necessary in any case to force sufficient air through a filter of moderate superficial area is so great that only installations provided with pressure fans can do the duty. Filters, too, after a certain period of use are liable to get choked up and diminish in efficiency, that is in permeability. It is therefore very desirable to provide a water gauge to indicate the difference of pressure between the two sides of the filter, and this gauge provides an indication of the time when the filter requires to be cleaned.

Wet Screens.

Another and perhaps superior method of cleansing the air from suspended impurities, and to some extent also gaseous impurities, is what is known as the wet screen method. This method consists of passing the air through a screen either of woven fabric, not easily subject to rot, or a screen of coke, pebbles, or other material, every part of which is kept continually wet by water trickling over it. The continual baffling of the air, and its necessary impingement on the wet surfaces brings about the almost complete removal from the air of suspended particles of impurity, and to some extent also of gaseous impurities.

The water is allowed to trickle down to the base of the chamber, whence it is generally run to drain, though in some cases where water is scarce, means are provided for circulating the water many times over the screen. The latter, however, is not a process to be recommended, as once passing over a screen is generally sufficient to make the water very foul and unhealthy.

Means must also be provided for periodical cleansing of such screens, either by removing and washing the material of which they are composed, or by replacing it by a fresh supply.

Similar results are brought about by the employment of an atomiser, which consists of a fine spray of water directed at a high velocity parallel to the currents of air on a small flat metal surface, fixed perpendicular to that direction, which causes the stream of water to be broken up into microscopic particles, which spread over the whole of the space through which the air passes, and thoroughly wets every portion of it, and presumably every particle of dust. The velocity of the jet of water should be not less than 60 feet per second, corresponding to a pressure head of, say, 60 feet of water column.

When this method is employed it is necessary that the air shall be subsequently passed over a system of baffle plates, which remove their superfluous moisture, which is always carried forward along with the air in the form of microscopical drops.

The disadvantage of this method is that it is liable to be abused, and that certain weather conditions make it easily possible to deliver the air to the flues in an almost saturated condition, or at least having a far higher degree of humidity than is desirable. It also requires a considerable amount of attention. The small hole through which the water flows is very liable to become choked up.

The washing of air previous to its being heated does not suffice for the adequate humidifying of the air, for even if the air is completely saturated at a low temperature, it does not contain anything like as much moisture as would be required to be provided with a suitable degree of humidity for a really well designed installation, so that the actual humidifying of the air must take place after it has been heated. Also for the reason that the amount of moisture which can be carried into the flue by saturated air depends entirely on the temperature of that air, and whereas the temperature of the air delivered is roughly constant, the temperature of the air coming in varies greatly, and cannot be relied on as a uniform moisture carrier.

CHAPTER VII.

THE MEANS FOR MOVING AIR THROUGH A ROOM.

NEITHER air nor any other body will move from one place to another unless there exists some force to make it move. In the case of a fluid this force takes the form of a difference of fluid pressure. Whenever a stream of air moves continuously from one point A to another point B, that fact alone is a proof either that the pressure at A is greater than the pressure at B, or that there exists sufficient kinetic energy in the air at A to force it along against a difference of pressure. The movement of the air from A to B will continue until the pressure at A and B are exactly the same or until the kinetic energy is exhausted.

If, therefore, a *continuous* stream of air is to be caused to pass from A to B, it is necessary that there should either be a higher pressure maintained at A than at B, or a continuous supply of kinetic energy. In either case this implies of necessity a continuous supply of power from whatever source derived.

Of course, air partakes of the nature of all other bodies, in that it follows Newton's second law of motion. If it is once set in motion it will continue in motion until it is stopped, either by some obstruction or other resistance.

If a continuous stream of air passes at a uniform velocity through any channel, or from any one point to another, then the total sum of the forces acting on the air, tending to make it move, is exactly equal to the total sum of the forces tending to make it stop. If, therefore, a continuous stream of air is to be led into a room, the pressure at the back of the current must be *maintained* greater than the pressure at the inlet to the room, otherwise the air will not continue to flow in that channel.

In making this statement, however, account must be taken of the static head, due to the weight of the various columns of air involved, as is explained more in detail in Chapter V.

Similarly, if a stream of vitiated air is to pass through an opening or up a channel, or otherwise out of a room, then the pressure inside the room must be greater than the pressure at the outlet of the channel, account having been taken of the difference of static pressure introduced by the weight of the various columns of air.

It is thus clear that the maintenance of ventilation in a room depends on the maintenance of pressure on the inflowing current greater than the corresponding pressure in the room, and also at the same time, the maintenance of a pressure in the room greater than the outlet pressure. The question of the means for ventilating a room resolves itself into the means of obtaining this descending scale of pressures, for it is quite certain that unless it is maintained no air will pass through the room at all.

It must also be observed that the pressure required for moving the air out of the room may be obtained from the means provided for forcing the air into the room, or separate means may be provided for extraction. Vice versa, the pressure required for the inlet may be derived from the means provided for the extraction. These three systems are called the plenum, mixed and vacuum systems respectively.

The means employed, operating either on the inlet or the outlet, or both, are

(1) What is called "natural ventilation" (whether or not assisted by heat).

(2) Mechanical ventilation, that is to say, means by which these differences of pressure are artificially maintained by moving mechanism, such as fans.

The "natural" methods depend entirely for their operation either

(*a*) On the velocity of the wind outside, which by means of suitable openings exposed in a certain way to the current of air induces a movement of air through the room, or in a channel or flue connected with such openings and with the interior of the room.

(*b*) The second method of bringing about the "natural" movement consists in maintaining a temperature in the room or channel, or both, different from the outside temperature, whereby the requisite difference of pressure is maintained, as will be presently explained.

Natural ventilation, by means of wind movement, being dependent on the velocity of the wind, is naturally very uncertain.

Ventilation by Open Window.

The simplest method by which the natural ventilation of a room can be effected is the provision of openings direct through the wall. In ordinary rooms these openings may take the form of windows, which can be opened or closed. It is probable that so far as the demands of hygiene alone are concerned, for the great majority of rooms this is by far the best method of ventilation at present known, especially in the British climate.

If there is a window at each end of a room, and if both of these windows are partly open, the degree of opening being suitable to the existing velocity of the wind, it is probable that the resulting ventilation is of a healthier type than any other yet devised. The only condition necessary to make this form of ventilation thoroughly satisfactory is that the occupants of the room should be accustomed to it, and that there shall be plenty of radiant heat in the apartment to give a feeling of comfortable warmth.

Such a method of ventilation, however, necessarily causes draughts, and there are very few people who have the perseverance to inure themselves to it, and there are probably few rooms in which the heating arrangements are sufficient to enable it to be used with comfort in winter. Besides the amount of opening of the windows needs continual attention with every change of the wind, otherwise the rate of ventilation will vary too much.

The healthiness of constant change of conditions has been alluded to in Chapter VI., and it is probable that this method secures that change in the easiest and most natural way.

It has, too, a striking effect on the amount of dust in the room when the building is in the open country or in pure air. In very dusty air probably the effect is to bring in a good deal of dust from the outside, which, although it is harmless (having been sterilised by the rain, sun, and wind), is nevertheless extremely troublesome.

Another great advantage of an open window is that the ventilation can be seen. It is not out of sight, and it is impossible for the occupant of a room to be ignorant of the fact that he has or has not any direct ventilation into the room. The opening is perfectly easy to keep clean; indeed it is kept clean as part of the ordinary daily cleaning that all well kept rooms receive.

But where a person is accustomed to having his windows closed, a concealed inlet is perhaps an advantage, in that such a person gets ventilation without knowing it.

The breeze through two windows arranged in this way will generally carry with it a large part of the dust in the air and will keep the apartment very largely free of such dust. This effect is often easily visible in sunlight. An immense number of dust particles can be seen floating about in the air of a closed room, which largely disappear on the opening of the windows. In no other form of ventilation does this occur.

Tobin Tubes.

Other inferior forms of this kind of ventilation consist of Tobin tubes, or other similar devices.

It is, of course, obvious that the popular impression that such openings in themselves produce any ventilation is absurd. They can have no positive motive power apart from the influence of the wind any more than so-called "ventilators" on a roof can. When air comes into such openings, if colder than the air of the room, it generally falls down on to the floor, and any person in the neighbourhood experiences a draught, which generally leads to the opening being permanently blocked up.

Ventilation by Vertical Flues.

The next form of natural ventilation consists of a vertical channel or flue of greater or smaller length which is connected with the interior of the room and terminates in an outlet exposed to the wind.

Apart from the effect which will presently be discussed, whereby the existence of a temperature in such a flue higher than that of the outer air, produces a suction in it, the exposure of the outside opening to a current of air almost always produces a suction in the flue bearing a certain relation to the kinetic energy of the wind, which relation depends on the surrounding conditions and on the shape of the outlet.

The outside air is rarely or never absolutely at rest. Even on the stillest day there is always *some* movement. The time-average velocity of the outer air is said by meteorological experts to be in the neighbourhood of four feet per second.

The magnitude of the suction pressure which can be produced by a wind of given velocity depends very largely on the presence and arrangement of walls and other obstructions in the neighbourhood of the outlet, and on the direction of the wind. Definite quantitative rules governing the amount of suction are impossible to obtain.

Cowls.

Efforts have been made by makers of cowls and similar appliances to make the action of wind on an open pipe more constant, by the introduction of baffle plates, fixed so as to catch and direct the currents of air passing by them. The degree of success attending these efforts has not been conspicuous.

No cowl, ventilator, or other such appliance, by whatever name it may be called, will work at all unless it is acted upon by a wind or by the application of heat.

In so far as the action of a cowl is intended to counteract the dynamic *pressure* generated by the action of the wind on obstructions in the neighbourhood of the outlet, all cowls are, and must of necessity be, quite useless.

Cowls, however, which are provided with non-return flap valves of mica, silk, or other light material, which close in such a way as to prevent any air from finding its way into the flue, can prevent actual down draughts in a flue due to this cause.

In so far, however, as the operation of the cowl or outlet depends on the *velocity* of the wind passing over the top, it is possible that suitably placed baffle plates may operate to prevent the *direction* of the blast from producing reversal of pressure in the pipe itself. For the baffle plates surrounding the opening may be so constructed that from whatever direction the wind reaches the outlet, the *direction* of the deflected current guided by the baffle plates may be roughly constant, and therefore the action of the cowl, in so far as it depends on the velocity as distinct from the pressure of the current, may be constant in sign and independent of the direction of the wind, though not, of course, constant in degree.

A horizontal wind passing over an open pipe is probably always more effective in producing suction in the pipe than if the pipe is covered with any form of cowl. A cowl probably always reduces the efficiency of an open pipe, and its only useful function in the case of a horizontal wind is, that it prevents rain, snow, etc., from beating down the pipe.

If, however, the cowl is fixed in a position where the direction of the air currents impinging on it is apt to vary considerably from the horizontal, then the cowl may serve a useful purpose in diminishing or preventing entirely down draught, but if it is fixed in a position such that the actual *pressure* of the air in its neighbourhood is liable to considerable variations, owing to the presence of high walls, trees, or other obstructions, then the only form of cowl which will be effective in preventing down draught is one including non-return flaps, which when they are acted on by any counter pressure, will at once close on their seats.

Pressure Cowls.

The cowl forms a means which is extensively employed under certain conditions for forcing air into a closed space, and less frequently for extracting air from a flue. They can only be employed with certainty of an effective result when a constant current of air can be depended on, and in the case of an extracting cowl when there are no obstacles in the neighbourhood which can cause the wind to generate a dynamic pressure.

The general form of cowl employed for forcing air into a pipe consists of a bent funnel, so arranged that it can always be turned to face the wind from whatever quarter it blows. In some cases the cowl may be mounted on a vertical axis as free from friction as possible and attached to a deflecting or directing plane, which automatically keeps it always facing the wind. This arrangement is very liable to get out of order, and is rarely used on that account.

The employment of ordinary cowls is chiefly effective in cases where an actual current of wind can be relied on, as for instance, in steamships, where the movement of the ship itself, relatively to the air, almost always produces a relative breeze, against which the cowl can be directed. There is, thus, always a certain velocity of air blowing down the cowl shaft, except where the velocity of the ship is the same as that of the wind in magnitude and direction. The same device is used for the ventilation of railway trains, and for other purposes. In such cases it is generally customary to reckon on a velocity of air being maintained in the cowl shaft of about 80% of the actual wind velocity outside unless the shaft is very long, or includes many bends or other resistances. In these cases the pressure due to the wind velocity can be calculated and the friction in the flues estimated. The velocity produced will be such as to absorb, by friction in the flue, all the pressure so generated. It is impossible to obtain precise figures which can be relied on in any given case, on account of the wide variations in essential conditions which necessarily exist.

Jets.

Other means for generating the slight difference of pressure for moving air, which may be regarded as auxiliary means for accelerating the natural circulation of air, consist of jets of high pressure air or steam. These operate by inducing a current in the space surrounding them. But as these jets always make a considerable noise they can only be applied in factories and similar buildings, where noise is of no account, or for producing forced draught for boilers or for similar purposes.

Columns of Warm Air.

The only other "natural" means for maintaining the difference of pressure necessary to produce constant, or moderately uniform, movement of air, is by maintaining a difference of temperature between the air in a vertical shaft and the outside air.

The principle, according to which this produces a difference of pressure, will be clear from the accompanying Fig. 9, in which A B is a column of air, say, one square foot in sectional area at the same temperature t° as the outside air. C D is a similar column of air, which is kept at a higher temperature than the column A B by means of a coil F, or other heating apparatus, fixed, say, in position shown. E is a closed valve shutting off connections between the two limbs.

Assume that thereby the temperature of the column C D is maintained at t_1° higher than t°. Now consider the balance of pressures at the point E. The pressure at the point B must be the same as the pressure at the point D, for both are open to the atmosphere.

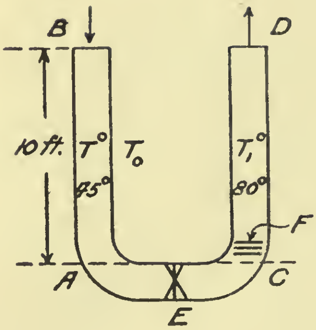


FIG. 9.

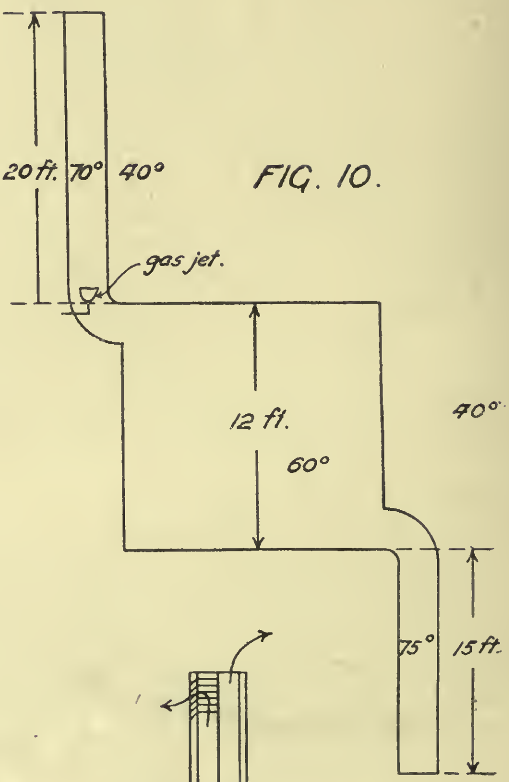


FIG. 10.

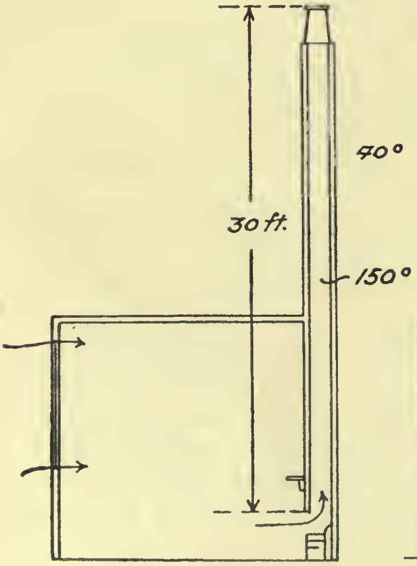


FIG. 11.

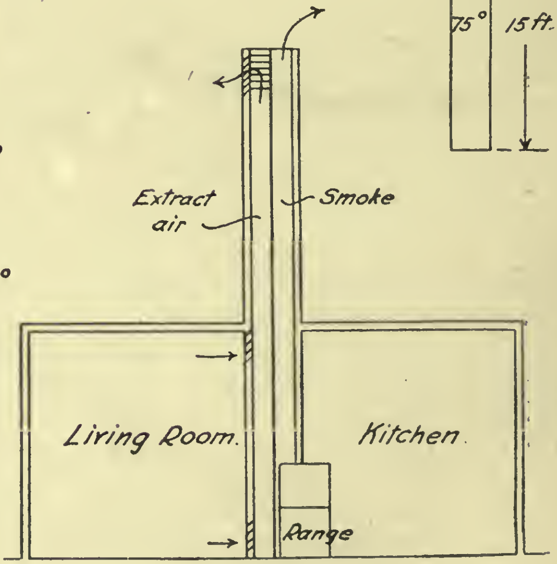


FIG. 12.

Owing to the weight of the column of air, A B, the absolute pressure at A will be greater than the pressure at B, by the weight of the column of air A B. Similarly, the pressure at C will be greater than the pressure at D by the weight of the column C D. But the weight of the column A B is greater than the column C D, because its temperature is lower. The air in A B is therefore denser than that in C D. The pressure on the two sides of the valve will be different. If the valve at the point E is now opened, there will be a continual stream of air downwards from B to A, and upwards from C to D, of such magnitude as to absorb this difference of pressure, partly by generating momentum in the air, and partly in overcoming the friction of the tube. This stream of air will be maintained the whole time the difference of temperature is maintained.

Now considering further the statics of this arrangement of U tube, it will be clear that since the temperature inside the column A B is the same as the outside temperature, the pressure at any level is the same inside the tube as outside, and this equality of pressure will be maintained as far as F, at the very bottom of the tube B A. It will not therefore make any difference to the pressure on the valve E if the tube A B is removed altogether, and the face of the valve is left open to the air.

If the valve is now opened and the same temperature is maintained inside the tube C D as before, the cutting away of the tube A B will improve the flow of air through C D, because the resistance of tube A B has not to be overcome. The air from the outside finds its way direct to the bottom of tube C D and passes upwards and out at the top.

The cylinder of air in the tube rises for exactly the same reason as that which causes a body lighter than water to float to the surface, namely, that the resultant hydraulic pressure on the under side of it exceeds that on the upper side of it by more than the weight of the body immersed.

The total pressure producing movement in this tube must therefore obviously be the same as before, namely, the difference between the pressure produced by a column of air of height A B at atmospheric temperature, and that of a column of air equal in length to A B, but at the warmer temperature in tube C D.

The magnitude of the forces acting on the air column may be calculated in the following manner:—

Suppose the column of air is, say, 10 feet high and 1 square foot in cross section, and its temperature is 80°, the temperature of the surrounding air being 45°. There is here a difference of temperature of 35°.

It will be seen from the Table V. that the weight of the warm column of air is 0.736 pounds. The weight of a corresponding column of air of the outside temperature, namely, 45°, is 0.787 pounds.

Suppose the absolute value of the atmospheric pressure at this level, A C, is 2,116.8 lbs. per sq. ft. At points higher up the atmospheric pressure will, of course, decrease by the weight of the corresponding column of air at 45°, so that the pressure at the top of the 10 foot column will be

$$2{,}116.8 - 0.787 = 2{,}116.013 \text{ pounds per square foot.}$$

If the warm leg, however, is closed at the top instead of the bottom, the table shows that the inside pressure at the top will be

$$2{,}116.8 - 0.736 = 2{,}116.064 \text{ pounds per square foot.}$$

There is here a difference of pressure between inside and outside at the top of 2,116.064 — 2,116.013 = 0.051 lbs. per square foot, corresponding to

$$0.051 \times 0.192 = .0098 \text{ inches of water at } 62° \text{ F.},$$

or

$$\frac{0.051}{0.0736} = 0.69$$

feet of air column calculated at the higher temperature.

It is thus clear that at the top of the warm column the pressure will be greater than the pressure in the surrounding air by 0.051 pounds per square foot, corresponding to 0.0098 inches of water column, or to 0.69 feet of air column of the same temperature, namely, 80°, as the warm column. All these figures may be at once found from the table.

If, therefore, a hole, *a* square feet in area is opened at the top of the warm column, air must at once commence to flow out of it at a velocity of

$$\sqrt{64.4 \times .69} = 6.7 \text{ feet per second,}$$

namely, the velocity which a falling body would acquire in falling from the top of an imaginary column of air 0.69 feet high. The sectional area of the stream of air which would flow out would be about 60% of the theoretical

$$0.6 \times a \times 6.7 \text{ cubic feet per second,}$$

owing, as has been explained, to the interference of the streams of air with one another.

Now the ventilation of a room is effected on the natural system by the application of heat in a manner which corresponds exactly with the explanation just given. Consider the case of a room which is provided with a vertical inlet flue, 15 feet high, coming into the bottom of the room, and an outlet flue, 20 feet high, from the top of the room, as shown in Fig. 10.

Assume that the outside temperature is 40° and the temperature in the inlet flue is maintained at 75°. Assume, further, that the room itself is maintained at a temperature of 60° by the inflowing stream of warm air, and that a coil of pipes, or gas jet, or some other means are employed to maintain the temperature of the outlet flue, which is 20 feet long, at 70°, or 10° higher than the temperature in the room.

Now the two flues and the room itself will form, jointly, a large irregular upcast flue, exactly similar to the flue C D on Fig. 9. But in this case the temperatures are not all uniform. The total motive pressure generated by the difference in temperature will be the difference in weight of a column of atmospheric air, equal in height to the distance from the bottom of the inlet to the top of the outlet flue, and the weight of a composite column of air at three different higher temperatures. A current of air will therefore pass through this room.

The method by which the air is distributed in the room itself is, of course, a separate matter, and will not be here taken into account, but it is assumed that the means for distribution are such that a uniform temperature of 60° is maintained throughout the room.

Ventilation by Fireplace.

Similarly, take the case of a room which is heated and ventilated solely by an open fire and a chimney of the usual type. Here the temperature of the chimney will probably be in the neighbourhood of 150° F., the height of the chimney being, say, 30 feet, and its sectional area 54 square inches. Here the total pressure available for forcing air up the chimney flue is equal to the difference in weight of a 30 foot column of atmospheric air at 40°, and a 30 foot column of flue gases at 150° F. (see Fig. 11).

In this case the entry of air into the room takes place through the walls, the windows and door cracks, and other accidental openings. There is no pressure tending to force the air into the room, except the excess of pressure of the outer air over the inner air, produced by the suction of the chimney flue.

Disadvantages of Ventilating by Warm Air.

The disadvantages of the heating of air as a means for promoting circulation of air through a room are considerable.

Cost.

In the first place, it is very expensive. Every thousand cubic feet of air at, or about, 60° F., which is raised unnecessarily through a temperature of 10° in order to promote ventilation, carries with it about 182 B.T.U.

There are cases where this heat can be secured at no cost, by utilising heat which would otherwise be wasted. Such, for instance, are the various methods of arranging the chimney flue from another fire in such a position as to warm the ventilating flue, as shown in Fig. 12. In such a case the heat required for warming the air for promoting ventilation is secured at no cost. This method has the disadvantage that the ventilating power ceases when the fire, whose waste heat is employed, is allowed to go out. If, for instance, the ventilation of a room is accomplished by the heat derived from the waste flue gases of the kitchen fire, the accelerated ventilation comes to an end the moment the fire is allowed to go out. If constant ventilation is required in the room, either an unnecessary fire must be maintained or a separate stove must be supplied which uses the same flues as its outlet, so as to maintain the temperature of the flue at times when a fire is not otherwise required. In the latter case, of course, the heat required for ventilation would not be obtained at no expense.

If the heat has to be specially supplied, as by gas jets or by steam coils, or otherwise, the expense may become really very considerable. One thousand cubic feet of gas, costing perhaps 3s., give off roughly, say, 550,000 B.T.U. This would suffice for the removal of perhaps 3,000,000 cubic feet of air under the above conditions, which are the most favourable possible.

The same work would be done even by a steam coil much more cheaply. The amount of steam which would be condensed in this process for heating the same quantity of air to the same degree, would be roughly 550 pounds. This might be obtained by the combustion of 80 pounds of coal, at a cost of perhaps 9d. The figures for a hot water coil would not be very different.

If, however, the same result were to be secured by means of an electrically driven fan, the total amount of electrical energy which would be required for operating the fan, so as to discharge the same quantity of air, would be about one unit or less, the cost of which would be about 1½d.

It is thus clear that, apart from their uncertainty, a very considerable waste of heat and therefore of money is caused by all systems of extraction by heating the extraction air.

The conditions are rather more favourable if the ventilating force is secured by warming the incoming fresh air. The heat employed in warming the air is then usefully employed in warming the room. But attention will be called in a future chapter to the disadvantages of this method of supplying heat to a room, and these must be taken into consideration.

Uncertainty of Heating by Warm Air.

A further serious disadvantage of the system of central ventilation of buildings, by means of heated flues conveying warm air to various points in the building without the use of a fan, is that such a system is very liable to reversal and to complete failure in certain states of the atmosphere.

If a room is ventilated by an inlet flue coming in at one side of a room and an outlet flue going out at the other, especially if the outer ends of the flues are at opposite sides of the building, the direction of the ventilation is liable to be wholly reversed when a wind blows on the building on the outlet side.

A comparatively moderate wind can easily generate far more pressure than can possibly be set up by a moderate difference of temperature in the flues. When it does this, if the inlet and outlet are in certain positions, the wind will blow down the outlet from the room and out of it at the inlet, thus not only carrying all the foul dust which is always deposited in outlet flues back into the room, but also sweeping out into the open air and wasting all the heat which is used for warming the inlet air.

This bad result can be avoided in some cases, at very considerable expense, by providing two sets of inlets, or two sets of outlets on opposite sides or corners of the room. But even so the device is clumsy, and the valves do not get operated at the right time.

Collection of Outlet Flues.

This trouble can also sometimes be overcome by collecting the outlet flues into a shaft in the ceiling space, the main outlet from this being carried above the top of the building.

This arrangement, besides being expensive, has the drawback that air sometimes syphons over from one room to another through the outlet flues, which are also liable to convey noises from one room to another. In the case of a fire occurring these communicating flues also facilitate the spreading of the fire from one room to another.

The difficulties attending the attainment of a uniform distribution of warm air through a complicated system of flues, deriving their heat from a centrally placed furnace, and their operative power solely from the temperature of the air, are therefore very great—so great indeed that many competent engineers shrink from undertaking the risk of failure involved.

The only certain way to distribute warm air through a complicated system of flues is the employment of a fan of sufficient power to overcome all tendency to irregularity.

With a centrally situated fan similar troubles may be found, if the fan is not of sufficient power, or is not run up to sufficiently high speeds to overcome all such variations as may arise, owing to differences in weather conditions.

In order that there shall be an equal distribution of air, even from a fan, it is necessary that the pressure maintained at the base of each flue shall be quite distinctly, in all atmospheric conditions, above the room pressure. In order to effect this result with certainty, it is desirable that the individual upcast flues leading to the individual air outlets shall be relatively small, and that the main flue shall be relatively large, and that the fan power shall be well on the top side of actual requirements.

If the proportion of the individual flue to the main flue is at all high, all the pressure of the fan may be lost at the few openings nearest to the heating chamber, and the remaining flues will then be in a similar condition to those which have been described already in connection with an air heating system, that is to say, there may be a possibility of air going into, instead of out of the openings. In order to avoid the possibility of anything of this kind, both the fan and the main channel should be of such capacity that the individual flues all taken together will not carry off the whole of the air which could be delivered by the fan at the given speed, and that the water gauge maintained in the central or main flue should be considerable, say from ¼″ to ½″. If these conditions are observed, constant uniform distribution of air can be relied on. This arrangement would, of itself, entail the disadvantage of flues which are too small to be easily cleaned. As easy cleaning is an essential condition of a satisfactory installation, it follows that the flues must be large and the resistance in them artificially introduced.

In order that it may be done, however, the fan must be of suitable power. This point is dealt with in the chapter on "fans."

Large Installations.

In the case of an air heating system of distribution, it is very desirable that the channels shall slope upwards, consistently, from the heating chamber itself. If it is possible a separate flue should be carried to every room. In this case there is little possibility of the short circuiting which is so liable to occur when the branch flues are taken from a common main trunk.

Regulation of an apparatus of this kind is an exceedingly difficult matter. It is possible to arrange a mixing chamber close to the stove, and a combination of dampers such that regular proportions of cold and warm air are introduced into the mixing chamber under suitable conditions, so that from the mixing chamber all the separate flues are led off to the various rooms. But even so the uniformity of working which is so desirable in a ventilating apparatus is not easy to obtain. Though there are, undoubtedly, many satisfactory installations of this kind at work, it is equally indisputable that there are many designed with the utmost care which are anything but satisfactory, owing to the peculiar conditions on the outside of the building.

It is frequently of value to have two fresh air flues leading to the chamber taken from different sides of the house. When this is done distribution of air is less likely to be disturbed by prevalence of wind in a particular direction, for the corresponding flue which keeps the best draught may be used in preference to the other one. The risk involved is, of course, that wind of a certain velocity may blow in such a direction that an intermittently recurring pressure is generated in the well or area from whence the air is taken. If the same wind tends to produce a greater counter pressure in some of the rooms to be heated by the warm air, then this wind must of necessity produce a reversal of the air current in the channels leading to the room.

Now if the air is taken from an area where the dynamic pressure due to the wind is higher, it may well be that this reversal of the current is stopped. But the application of air heating to any system of flues of anything like a complicated character is so difficult that it is only to be recommended in particular cases.

Fans.

The standard method of moving considerable quantities of air is by a fan either of the propeller or the centrifugal type. This is the only method whereby a certain definite and large interchange of air can be secured in all states of the weather, and in all circumstances. All other methods are liable to be uncertain and to be injuriously affected by particular states of the atmosphere and prevailing winds, etc.

The fans used for air propulsion are, broadly speaking, of two classes, the propeller type and the centrifugal type. The former type are efficient for moving large volumes of air against very low resistances, but become very inefficient when the resistances to be overcome are considerable. The limiting pressure against which the propeller type fan should be employed should not be more than 1/16″ to 1/8″ of water gauge. They are therefore unsuitable for forcing air into long or sub-divided flues, and are chiefly suitable for short flues, and for positions where there is no flue at all.

These ventilators are not constructed above about 8 or 9 feet diameter, the peripheral speed at which they should be run should not be above about one mile per minute, or roughly, 100 feet per second, otherwise they will produce a very penetrating humming noise. Accurate information in regard to the performance of fans of this type is very difficult to obtain, owing to the essential difficulty in estimating the amount of air which they deliver, and owing to the fact that slight differences in design produce considerable differences in performance. Reference is made to this in a special chapter on fans.

A fan which is worked by strap or rope is generally less noisy than a direct coupled fan.

Centrifugal fans are made in large variety of types.

In order to avoid noises the peripheral speed should not be greater than about 150 feet per second, which generally corresponds to a water gauge of about ⅝″.

The choice of a ventilator of suitable size and type is generally made to some extent on commercial grounds. If the best value is to be obtained from the money expended, regard being had to the subsequent cost of maintenance, it will generally happen that a fan relatively expensive in first cost must be employed. In cases, however, where (as is unfortunately too frequent, especially in this country) the placing of an order is dependent almost wholly on first cost, the engineer cannot always afford to take into consideration subsequent cost of operation.

It will be seen from the tables given in the chapter on fans that the cost of maintenance for power, etc., for a small or inefficient fan may easily be double of what it would be for the same output with a larger or more efficient fan. The difference in cost of the two fans might be more than repaid by a single year's difference in cost of upkeep.

In order to make a proper selection of the type and size of fan required, the heating engineer must know with some exactness the following particulars :—

1. What is the total quantity of air to be passed by the fan?

2. What is the pressure against which this quantity of air must be forced?

3. What power is required to work the fan in order to deliver the requisite quantity of air at the requisite pressure?

4. What would be the effect on the consumption of power of diminished or increased output to suit particular circumstances?

5. What is the temperature of the air in which the fan will work?

It is of the greatest possible importance for an accurate choice of fan that the engineer should be provided with curves, similar to those given in the chapter on fans, which will supply the necessary information.

To review the various means for moving air through a room, they may be classified as follows :—

1. Cowls or hoods which are only effective if a constant wind or air velocity can be relied on.

2. Heating of the air in upcast shafts.

3. The employment of fans.

CHAPTER VIII.

ON THE MOVEMENT OF AIR IN A ROOM.

THE problems of ventilation have been indicated in general terms in the previous chapter. The ventilation of a room does not consist, as so many people seem to think, of passing fresh air into a room and thereby forcing the vitiated air out, much as a piston in a cylinder might force exhaust steam out before it, impelled by the pressure of live steam behind it. It would in many ways be a great advantage if ventilation were such a simple matter as this, but unfortunately it is not. Nor can it ever be made to resemble this action in any real way, although some systems of ventilation appear at first sight to produce the result more nearly than do others.

Upward Ventilation.

One might imagine at first sight that the ideal system of ventilation would be brought about by forcing air at a suitable temperature into a building all over the floor at a very low velocity, whereby the vitiated air vapour, etc., given off from the bodies of the persons in the room would be slowly displaced upwards out of the breathing plane to the ceiling of the room or hall, from which it could be carried off by suitable channels and fans. This system is diagrammatically indicated in Fig. 13. It is quite conceivable such a system could be carried out in practice in such a way as to give very good results, *provided it received sufficient attention*. But the practical difficulties in the way of its realisation are so great that only in very few cases is it possible to carry it out in a satisfactory manner. The expense of the construction is great, as is also the amount of attention required in order to satisfy at the same time the demands of hygiene and comfort.

If the air is introduced through small holes in the floor, the result may be very bad, because of the dirt which must inevitably find its way into these holes, and through them into the air channel beneath, from the boots and bodies of the persons occupying the room. In the case of a miscellaneous crowd, the character of the dirt which would certainly find its way into the fresh air channels would be of a very offensive description.

The only way in which air could be introduced, in order to avoid this possibility, would be at a short distance above the actual floor level, through outlets constructed in such a way that while the air could get out into the room dirt could not easily get into the flue.

The only way in which this can be effected is by raised inlets under fixed seats, as indicated in Fig. 14. It is, therefore, in some cases suitable for such buildings as churches, theatres, concert halls, where the seats are fixed, and indeed it has been sometimes carried out in such buildings. The expense, however, is great. The necessary construction demands that the whole of the space under the floor shall be one large air chamber, or a number of large channels shall be constructed leading to the various openings.

In order that such a system may be maintained satisfactorily it is necessary that all the openings and flues leading to them should be regularly and thoroughly cleaned. The system has the grave disadvantage that objectionable smells, vapours, etc., are often given off from the boots, clothes, or bodies of the occupants, and that all the most dangerous portions of the dust (which exists everywhere in an enclosed space occupied by living beings) tends to accumulate near the floor. This method of introducing fresh air carries all these objectionable elements directly into the breathing plane.

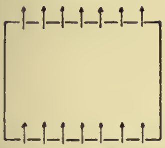
In any case, it is only with fixed seats that such a method of distribution is practically possible, and as a large number of halls and large buildings are not provided with fixed seats, the system is impossible to carry out without perforating the floor, which, for reasons of hygiene, is altogether undesirable.

Another serious disadvantage of the system is that if the temperature and velocity of the incoming air are not maintained within narrow limits, it gives rise either to complaints of cold feet, or, if the air is introduced very slightly warmer than the general air of the room, to complaints of "heat fumes" and stuffiness.

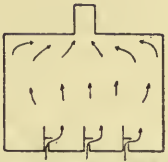
Downward System of Ventilation.

The exact reverse of this system, shown in Fig. 15, has in some respects great advantages, although it may not appear so theoretically perfect.

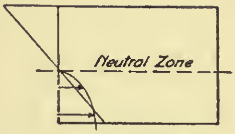
If the fresh air is forced in all over the ceiling of a room, the top layer of fresh air so introduced may be conceived as forming a kind of piston,



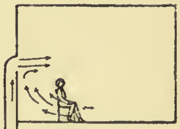
Ideal upward ventilation.
FIG. 13.



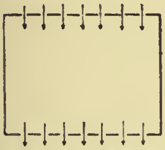
Practical upward ventilation.
FIG. 14.



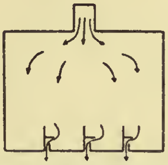
Primary draught.
FIG. 23.



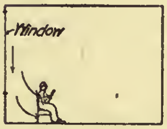
Secondary draught.
FIG. 24.



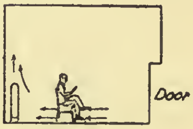
Ideal downward ventilation.
FIG. 15.



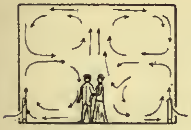
Practical downward ventilation.
FIG. 16.



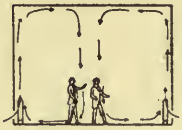
Condensation draught.
FIG. 25.



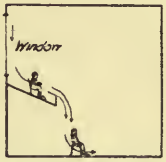
Induced draught.
FIG. 26.



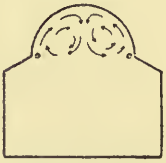
Warm ventilating radiators.
FIG. 17.



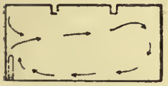
Very hot radiators.
FIG. 18.



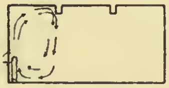
Condensation draughts in church.
FIG. 27.



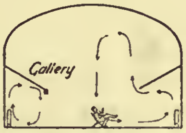
Church with overhead pipes.
FIG 28.



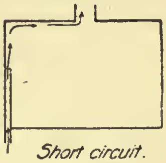
Warm radiator.
FIG. 19.



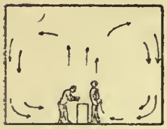
Very hot radiator.
FIG. 20.



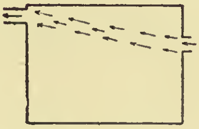
Effect of heat under galleries.
FIG. 29.



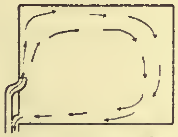
Short circuit.
FIG. 30.



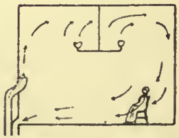
Central Stove.
FIG. 21.



Short circuiting current.
FIG. 22.



Best distribution of warm air with single inlet & outlet.
FIG. 31.



Bad effect of downward ventilation with gaslights.
FIG. 32.

DIAGRAMS SHOWING DIRECTION OF AIR CURRENTS IN A ROOM.

which slowly forces the lower vitiated air out through openings in the floor, or under the seats, and so carries it away.

This imaginary piston-like displacement, however, does not even in this case exist in any real sense. The air exhaled by the occupants, and the warm vapour emanating from their bodies, actually rise immediately on their liberation, in virtue of their high temperature and of the upward currents generated by the heat of the body, and become mixed with the descending cooler and fresher air before being expelled through the floor openings. The chemical composition of the air which is expelled does not therefore differ greatly from that of the air which is being breathed, except in regard to its contamination with dust.

In the opinion of the present writer the chief distinction in effect between the upward and downward systems of ventilation, when carried out by forcing air through sub-divided floor inlets or ceiling inlets respectively, consists in this, that the upward system tends to decrease the proportion of CO_2 and excreted vapours, and increase the proportion of *dust* in the breathing air, while the downward system decreases the dust and increases the carbonic and vapour impurities.

Even in the case of downward ventilation it is not possible to have openings in the floor itself, if proper regard be had to hygienic considerations. Such holes, as in the previous case, would speedily become very foul, and no caretaker, however careful, could be expected to give individual attention to each hole, such as would be necessary if the whole installation were to be kept in a proper hygienic condition.

An installation of this kind would accumulate organic dirt on the edges of the individual holes, which would putrefy, and as soon as the ventilation was stopped the gaseous products of putrefaction would be felt in the hall as very objectionable, so that even in the case of downward ventilation fixed seats must, in practice, be employed having raised fixed openings under them, as shown in Fig. 16.

The difficulty of cleaning thoroughly under fixed seats where there are no obstructions is great enough, and this difficulty is multiplied by the provision of a large number of short narrow flues projecting upwards from the floor.

It is possible, of course, to form these flues as part of the standard supporting the seats, especially where these are arranged in the form of benches; but in most modern buildings of this character, if provided with fixed seats, the latter take the form of separate chairs for each occupant, and the difficulty of cleaning under them is greatly increased by a number of flues passing through the floor.

In any case, it is necessary that the *interior* of each inlet should be periodically and thoroughly cleaned, and this could in most cases only be done from below, unless the seats were of a special design.

One of the chief points to be aimed at for satisfactory ventilation on any system is a thorough cleansing of all the parts with which air can come in contact and the complete removal of all matters of an objectionable nature, which, of necessity, are introduced by the occupants.

Introduction of Fresh Air at the Walls.

In a room with a solid floor the fresh air, if not introduced through apertures in the ceiling, must be introduced round the walls. This at once precludes the possibility of any such piston action, as has been described above, from taking place.

Given that the air can only be introduced round the walls, and that the persons breathing the air are occupying the centre of a room, it is quite obvious that the only way in which fresh air can reach the occupants is as follows :—

The occupants give off from their persons, and from their lungs, vitiated air and vapour, the character of which has been already discussed in Chapter VI., so that each occupant is surrounded as it were by an invisible halo of foul air and vapour, which must be removed or diluted by the admixture of fresh air. This halo gradually spreads itself upwards from each individual by diffusion. An analysis taken of the air at different points in a closed room or hall, outside a distance of perhaps 24″ from the body of the persons concerned, shows a certain degree of uniformity, unless special conditions are present. Within that distance it is probable that the actual waste products proceeding from one individual could be detected by analysis.

It is therefore clear that the actual process of ventilation within a room, where the fresh air openings are on the wall, consists of the carrying of fresh air by horizontal air currents, and by diffusion from the points at which it is introduced, all over the room. This system is diagrammatically indicated in Fig. 17.

What actually takes place within a ventilated room is that fresh air is introduced at one or more points, that air presumed to be foul is taken off at one or more points, and that the distribution of the fresh air and the collection of foul air are both effected by means of irregular convection currents within the room itself, and that the average composition of the air in the breathing plane cannot be ascertained with certainty and exactitude by an analysis of air taken at one particular point.

If a sample of air were taken from a particular point in a room which happened to be in the path of a current of fresh air, proceeding from the fresh air inlet, it is quite conceivable that it would show on analysis a very high degree of purity, although the actual average condition of the air in the room might be extremely bad.

Instances frequently come under observation in which a direct current of fresh air is detected in a badly designed ventilation installation from the inlet direct to the outlet, as shown in Fig. 22. The incoming air, if introduced at a high velocity, is sometimes known to form a tunnel for itself through the circumambient layers of vitiated air, so that even a sample of air taken in close proximity to the outlet might reveal a composition much superior to that of the average room air.

The opposite phenomenon is not so likely to occur, that is to say, it is hardly likely that any analysis of air, taken at a point in a hall or room not very near the outlet for vitiated air, would reveal, on analysis, a composition much *worse* than the average composition of the air in the room, unless such sample were taken in the immediate neighbourhood of a human being, in which case it is conceivable that a large proportion of the sample taken might consist of breath which had been exhaled immediately before the sample was taken.

On the other hand, if the sample were taken near the outlet to the extraction shaft, it would probably (in the absence of any short circuiting current) reveal a composition slightly worse than the average composition of the air in the breathing plane, especially if the extraction outlet were situated at a high level.

There is, of course, a tendency for vitiated air to rise, owing to its temperature, in spite of the fact that, calculated on composition alone, it is specifically slightly heavier than normal pure air. The CO_2 in its composition tends to make it heavier, but this effect is almost counterbalanced by the increased proportion of moisture, which is lighter than air.

These considerations are sufficient to show that the actual ventilation of any room consists of an extremely complicated system of air currents of different degrees of vitiation. Every part of the room, every particle of the heating surface, every pipe, every gas jet, every human being, every inlet, every outlet, every crack in the walls, almost every square inch over the entire surface of the walls, produces its own individual effect on the movement of air currents in the room.

Now what is actually aimed at for the efficient ventilation of a room is to produce the best possible result *all over the breathing plane,* which may be fixed at about five feet above the floor. What is desired, in fact, is so to control the inevitable air currents passing through the room that an average good result shall be secured over as much of that space as possible, and that system of ventilation is the best which produces this result.

What is a true criterion of the value of the result, whether it is to be judged by chemical, hygrometric, or bacteriological composition, by temperature or in what other way, is a matter for the medical authorities to decide. There appears at present to be no self-consistent information available on this matter.

Unfortunately, however, this is not the only consideration which the heating engineer must have in mind.

The public does not judge a system of ventilation on the analysis of the air, but with regard to the feeling of *comfort* which it produces. If it were possible, as it is sometimes possible, to produce a feeling of comparative comfort in a hall of which the air is really very bad, such a hall would acquire the reputation of being very well ventilated.

In the present state of public education the chief feature of a ventilating arrangement, which nine people out of ten regard as important, is that in no part of the hall shall they be conscious of any movement of air; such a perceptible movement of air is called a "draught," and by that name is condemned.

On Draughts.

A draught may broadly be described as a perceptible current of air. It may be taken as an axiom that such a current of air impinging on the body of any person will give rise to complaint.

It will be well, therefore, to consider the causes of all such draughts, and how they can be avoided.

They may be divided into four classes, which are not, however, absolutely distinct from one another—

(1) Direct or primary draughts of inflowing fresh air.
(2) Secondary draughts, caused by induction from the primary draughts.
(3) "Condensation" draughts, caused by the cooling of air in contact with the colder bounding surfaces of the room.
(4) Induced draughts, caused by the upward current generated by radiators or other heating surface.

Instances of all these types of draught are illustrated in the diagrams, Figs. 23 to 26.

Draughts under category (1) are caused directly by the position of the neutral zone in the hall or room (as to which, see Chapter X.). If this is at a high level every crack or cranny through the windows, walls, doors, or floor, gives rise to a draught, for the outer cold air will get in wherever it can.

These draughts can only be suppressed, either by depressing the neutral zone (see Chapter X.), or by stopping up the holes through which the draughts come. They may sometimes be rendered imperceptible by warming the incoming air.

Thus, draughts finding their way into a church through a cold lobby are frequently rendered unobjectionable by warming the lobby. The draught is not thereby suppressed, but the incoming air is warmed.

Draughts in category (2) do not exist except in cases of forced ventilation, especially on the plenum system. If air at a considerable velocity is introduced into a standing mass of still air, it naturally induces in it secondary currents, and any person sitting in the path of one of these currents will feel the movement of the air as a cold draught, even though the actual temperature of the moving air is the same as, or slightly greater than, the temperature of the air surrounding him (Fig. 24).

Draughts such as this can be cured by the provision of screens or baffles, which divert the position or direction of the induced stream, so that it does not pass over positions which will be occupied, or by reducing the velocity of the incoming stream, or altering its direction, or increasing its area, or in some similar way.

The draughts which are generally the cause of complaint are probably those in category (3), and are caused in large rooms, halls, and the like, by the cooling of air in contact with the ceiling, windows, and walls. The cooled air becomes denser than the rest of the air in the room, and falls down the walls to the floor, where it is diverted into a horizontal direction and impinges on persons seated near the wall (Fig. 25).

These draughts can in most cases be cured by the provision of suitable heating surface underneath the place where the draughts are generated. The joint effect of the down-flowing stream of cooled air, and the up-flowing stream of warmed air from the radiators, or other surface, is that a horizontal stream of air is generated overhead where the two streams meet, which may be very valuable for distributing fresh air from behind the radiator. This is shown in Fig. 17. If there is no fresh air inlet behind the radiator, it is desirable to place a sheet of metal behind the radiator which is bent over the top of the radiator. This leads the part of the current which clings to the wall to the back of the radiator, and so warms it before it reaches the floor, besides preventing the upward current from the radiator from blackening the wall above.

In large rooms, churches, halls, etc., such draughts are commonly caused by the cooling of the air in contact with the ceiling, large windows and walls. If such buildings are surrounded by a gallery, the cool air flows from the roof down the walls, and being stopped in its descent by the gallery, pours in a cascade over the edge of the gallery on to the people sitting below, giving rise to discomfort all along its path. This is shown in Fig. 27.

This trouble can be cured by the provision of heating pipes at the back of the gallery. Sometimes also pipes are run round the edge of the gallery itself.

Draughts due to the cooling of air in the roof are regularly cured by the provision of heat at a high level, either by steam pipes or gas burners, or other source of heat. This causes a local circulation of warm air near the ceiling which draws into itself the air cooled by the roof, and prevents its descent into the zone of the people present (Fig. 28).

All high buildings should be provided with a certain proportion, generally from 20% to 25% of their total heating surface at a high level, generally about half the total height of the room. If this is done, down draughts from the roof will give little trouble.

The draughts in this and the next category form the principal means whereby heat and fresh air are distributed in a room.

(4) Such draughts as this can only be overcome by a careful consideration of the heating effect of the heating surface.

For instance, if a powerful radiator is placed immediately opposite the door leading into the warm room from a cold passage or hall, the upward current generated by the radiator will sometimes draw to itself a current of very cold air coming in through the door cracks, and any person sitting in the path of it will experience a cold draught past his feet. Such draughts as this can be overcome by baffles or screens, and the like.

A careful consideration of the effect of the heating surface placed in large halls should always be made. Each radiator or pipe coil should be considered as causing an upward flow of air. A consideration of the shape of the hall, and especially of the cold parts of the wall, should indicate where down draught is likely to occur in a room containing radiators fixed at various points. On the walls the down draught can only occur between the radiators. Quite a small amount of heating surface will generally be sufficient to counteract it. Two, or even one length of 2″ pipe fixed in the window sill of a large window will often cure completely the tendency to down draught.

The Effect of Windows.

The temperature of the air underneath a large window is normally from 3° to 5° below the temperature of the room. The loss of heat from every square foot of a single window is about one B.T.U. per square foot, per degree difference of temperature, per hour.

A window, say, 20 feet high by 10 feet wide, containing 200 square feet will therefore cool about

$$\frac{200 \times 30}{.0764 \times 3 \times 0.237} = 111{,}000$$

cubic feet of air through three degrees per hour. The depth of the draught from such a window may be taken as about 18 inches. The mean velocity of air which may be expected at the bottom will therefore be about

$$\frac{111{,}000}{3{,}600 \times 15} = 2 \text{ feet per second.}$$

In order completely to suppress this draught would require a radiator below, which at a total emission of 140 B.T.U. per square foot should have a surface of

$$\frac{6,000}{140} = 43 \text{ square feet.}$$

But the draught may be reduced below two feet per second by a much smaller radiator than this, and if this is done there will probably be no complaint of draught at all.

Each window thus produces a down draught of an approximately calculable velocity, and it is possible to anticipate with some accuracy the movement of air in each part of a room if all these matters are taken into account, but such estimates are liable to be seriously upset by a wind acting on the outside.

The effect of a wind acting on the surface is that cold fresh air is introduced all over that surface, through all the pores in the walls, and through every window crack. An estimate may be made of the quantity of air thus finding its way into the room by means of the figures given below, but such estimates are liable to be widely in error, howsoever carefully they are made.

The quantity of air passing through window cracks has been the subject of experimental investigation by Mr. H. W. Whitten, of Boston, Mass., and the results may be found in the proceedings of the American Society of Heating Engineers, Vol. XV.

Broadly speaking, the result is that the clearance of a sash window is generally 1/16", and the amount of air which must be reckoned on as leaking in through such a crack is about 150 cubic feet per hour per lineal foot of crack for a 15 mile per hour wind. The quantity varies as the square of the wind velocity.

In order that the ventilation of a room may be satisfactory from the point of view of comfort, it is in almost all cases very desirable that the heating shall be in full work for some hours before the people assemble, for during the heating up of a cold room the draughts generated are always considerably greater than after the room has been heated for some time. This is owing to the low temperature of the walls, etc., which causes great differences in the temperature of the air in their immediate neighbourhood.

Distribution of Fresh Air.

It is by the air currents generated by the heating surface that both heat and the fresh air are distributed to all parts of the room. According to the results of some experiments and observations made by the present Author, the following is the explanation of the general principles on which the distribution takes place.

Where there are a large number of persons assembled in a hall the heat given off from their bodies has a tendency to cause an upward current of air in the centre of the room. This necessarily implies a horizontal stream of air to take the place of the rising air.

If there is no heat round the walls the air which is drawn to the centre of the room consists of the air previously cooled by the action of the walls and windows; owing to its greater density it flows along the floor, giving rise to complaints of cold feet. This is shown in Fig. 21.

Where radiators are fixed round the walls, however, the upward current which they cause, meeting the downward current from the walls and windows, produces a resultant current of tempered air at a level above the heads of the people present instead of along the floor, as shown in Fig. 17. This current also draws into the circulation some of the air near the floor. Thus, in a room heated by radiators round the walls, the direction of the air current at the floor level is normally *outwards* towards the radiators, whereas with the same room heated by radiators fixed in the middle of the room, the direction of flow near the floor is from the walls inwards. This is shown in Fig. 21.

It is this same action which causes the distribution of fresh air from a ventilating radiator, as a radiator which is provided with a fresh air opening behind it is improperly called.

The induced upcast produced by the radiator, combined with the height of the neutral zone (as to which, see Chapter X.), draw in the fresh air from the outside through the opening, and at the same time the upward current draws in to itself air from the middle of the room. Above the radiator the mixture of these streams of air takes place, and the massed stream is again mixed with the cooler air which is flowing down the walls above the radiator (see Fig. 17).

In order that the pressure in the room may equalise itself, this abstraction of air from the centre of the room is made good by horizontal currents taken from the rising current, both at the ceiling and floor levels. The existence of a horizontal current towards the outside walls at a high level, towards the centre of the room at a mid level, and from the centre of the room towards the radiator at a low level, can be easily detected by a sensitive anemometer. It proves the existence in a really well ventilated room of four separate vortices of air, a pair above and a pair below, as shown in Fig. 17.

Effects of very Hot Radiation.

The greater the temperature of the radiator the higher is the position of this middle inward horizontal current. With a very hot radiator in a relatively low room, the horizontal current is just underneath the ceiling, and suppresses the normal outward current at that level. The pair of lower vortices thus expand and monopolise the whole space, suppressing at the same time the upward current from the occupants and bringing down in place of it into the breathing plane a current of overheated air charged with impurities, which radiators at a lower temperature would allow to circulate harmlessly overhead (see Fig. 18).

This is one of the disadvantages of a very hot radiator, because the ceiling is almost always the region of the greatest vitiation, and if the vertical current reaches this region before taking the horizontal direction, the down coming air in the centre of the room is more vitiated than it is where the horizontal stream is lower, owing to the temperature of the radiator being lower.

A very hot radiator also produces a great difference of temperature between the ceiling and the floor, which gives rise to complaints of cold feet and hot heads, which is a common complaint against heating systems. The explanation of this is, in the opinion of the Author, to be found in the above fact.

The distribution of heat from a very hot radiator takes place by air currents along the ceiling. Therefore, if a long low room is heated by a radiator fixed at one end, care must be taken that there are no projecting beams or other projections in the ceiling which will prevent or deflect the warm air currents which effect the distribution of the heat.

A single beam across a ceiling will cause the circulation to be local and largely confined to the space between that beam and the radiator. This effect is shown in Fig. 20. This space may easily be overheated, the more distant portion of the room being relatively cold. The removal of the beam or a drop in the temperature of the radiator would cause the temperature to be more or less uniform throughout the room, as shown in Figs. 19 and 20.

For this reason it is necessary that if the ceiling of a room is divided up in this manner by any partitions or beams, or other similar structures, a separate radiator corresponding to the requirements of that section must be placed in each section, otherwise uniformity of temperature will not be secured.

The disadvantage of very hot radiation is very marked where the room is lighted by gas jets. For the hotter the radiator the more tendency there is to draw down the air vitiated by the gas jets into the breathing plane. The whole of the space above the level of the gas jets must always be regarded as the region of vitiated air, and where a building is lighted by gas, no tolerable system of ventilation other than the upward is possible, for any effect which tends to bring the products of combustion down into the breathing plane must be avoided. This is illustrated in Fig. 32.

Galleries.

In the case of a church surrounded on all sides by a gallery similar effects are to be observed.

If there is a considerable quantity of heating surface under a gallery a local circulation will be caused, as shown in Fig. 29, left hand side, which will overheat the space under the gallery, while having relatively little effect on the body of the air in the centre of the hall, especially where the ceiling underneath the gallery is bounded by a beam.

Where there is no beam the rise of the warm air past the edge of the gallery tends to cause a down draught in the centre of the church, unless a substantial amount of heating surface is provided at a higher level (see Fig. 29, right hand side).

A useful system of ventilation can sometimes be used in which alternate bays under the gallery are utilised for distributing fresh air to the centre of the church, for drawing off warm air at the back of the gallery, and delivering it above the gallery.

Separate bays under the gallery are sometimes partitioned off by girders, so that there is no direct flow from the inlet bay to the extract bay.

In this arrangement, of course, it is inevitable that the vitiated air generated under the fresh air bays of the gallery shall mix with the incoming fresh air, and the people in the centre of the church only receive partly vitiated air.

Cooling of Rooms.

In all cases where it is sought to cool a room by the introduction of cooler air, the most important point is that this air shall be very finely subdivided before it is distributed. The most modern practice is that all such air shall be introduced from the roof or ceiling through a very large number of openings, and that it shall be discharged into the hall in a horizontal direction, radiating from the centre of the openings, so that there is little or no initial velocity downwards, and the separate air streams neutralise one another before they begin to move downwards. Then there is no tendency for cold fresh air to gather in a large mass, and there is no tendency for a draught to form in one spot rather than in another.

If, however, a large layer of cool air exists in the neighbourhood of a large window, or other cooling surface, the existence of this window is sufficient to *commence* or give the lead to a draught, which may be a very serious one, as the cold air tends to follow the lead of a draught once it is commenced.

In some cases, therefore, it is necessary, while cooling a room or hall by means of the introduction of cold air at a high level, to warm a large window or other cooling surface, so as to prevent a large part of the cold air from forming a severe draught. It is necessary to repeat that such an arrangement as this cannot possibly be carried out in rooms lighted by gas.

The best installations abroad are provided with entirely separate plant for warming and cooling. This, of course, doubles the expense and is only to be thought of in cases where the best result is to be produced, regardless, not only of cost, but of space, that is, where the building can be designed round the ventilating apparatus, rather than in the condition where, as is usual in this country, the ventilating plant has to be squeezed in where and how it can.

Frequently a very carefully designed installation may be rendered useless by bad management, and the more complicated and expensive a ventilating arrangement is, the more is this danger to be feared.

If, for instance, a hall is carefully designed for different operation during mild weather, and during severe weather, and if the part of the installation designed for the severe weather is used for the mild weather, or vice versa, the results are likely to be more serious the better the design of the installation.

In keeping down the temperature of a hall it is always easier to prevent it from rising too high than to reduce it once the temperature has risen. It is therefore advisable to commence the cooling operations when the temperature is several degrees below the maximum allowable. The temperature will continue to rise, but more slowly, and may be kept within the given limits, which might well be impossible if the cooling were not commenced until the maximum had been reached.

Rooms Lighted by Gas.

Wherever a large hall is lighted by gas jets, one of the first things to be considered by the ventilating engineer is the means for withdrawing the fumes from these gas jets from the air before it mixes with the body of the air in the room. This provision is frequently expensive, and if it is attempted to do it in an inexpensive manner, the result is frequently very ugly.

Where gas rings, or sun burners, and the like, situated near the ceiling are the sole means for lighting a large building, they may be made a very efficient engine of ventilation, but only where the ventilation is on the upward system.

In this case, of course, rising trunks and extract flues are provided in the roof, and the powerful upcast in these is maintained by the heat from the gas jets.

The flues for carrying off these products should in all cases be of metal, and they should be provided with closing dampers, which can be operated from below by ropes or chains passing over pulleys.

If they are left open when no gas is alight, in certain states of the wind large volumes of cold air may come through them, which falling direct into the body of the building produce really serious cold draughts.

The difficulty in this case is only in the adequate distribution of the fresh air. The means by which this distribution may be effected has already been dealt with.

The upward system of ventilation is always more expensive in upkeep than the downward system to produce the same result, because with the upward system the hottest of the air is removed, whereas with the downward system the coldest is removed.

Introduction of Fresh Air.

The position for the introduction of fresh air into an occupied room, especially one which has a large number of occupants, must always be carefully considered.

It is rarely possible to change the air of a large hall more than four or five times at most per hour without the production of noticeable draughts. It must be assumed that in cold weather all incoming fresh air must be warmed before introduction into the room, otherwise primary draughts are almost unavoidable.

Even warm air must not be allowed to impinge on persons sitting near the inlet, otherwise it is certain to give rise to complaint of cold draughts.

These complaints will also be made if the air is introduced at any considerable velocity, owing to the secondary or induced currents.

For the introduction of a considerable volume of air the best way is to discharge it vertically up the wall at a height of about six or eight feet above the floor through channels formed in the wall or behind the dado. Care must, of course, be taken that these channels can be easily cleaned.

Air, however, introduced as through the old Tobin tubes at such a height as this always causes a cascade of cold air, which makes the neighbourhood of the inlet intolerable, as previously stated. In many or most cases Tobin tubes are either kept closed or are permanently blocked up.

Care must be taken that air is not discharged vertically up a wall which is likely to show markings or streakings. It is inevitable that if a constant current of fresh air passes up a wall, the wall will be blackened, however clean the air may be made. It is not necessarily the dirt originally in the air which causes this streaking, but the induced current carries into the stream dust from the room itself, which is gradually deposited on the walls and causes its blackening.

If the air so introduced is considerably warmer than the air of the room, especially if the velocity is considerable, the current of air clings to the walls until it reaches the ceiling or gallery, or other horizontal surfaces. The current then breaks up and spreads itself over the horizontal surface, as shown in Fig. 30.

As named above, where a room or hall is lighted by gas jets there will always be a layer of highly vitiated air at the ceiling, and it is important in this case that the main fresh air current is not allowed to reach this vitiated layer, otherwise some of it will be drawn into the breathing plane. Thus, with a room lighted by gas jets it is extremely undesirable that the introduced fresh air should be at a temperature considerably higher than the temperature of the room. Indeed, great care is necessary in the ventilation of such rooms as this (see Fig. 32).

On the other hand, even though there are no gas jets, but if the ventilation is upward, the extraction air being taken from the ceiling by an extraction fan, the fresh air if introduced too warm will probably reach the outlet opening before it has had a chance to mix with the room air. Thus, when warmed air is introduced in this manner in a room having upward ventilation, the method is very wasteful. Indeed, it is not too much to say that two-thirds of the fresh air introduced may easily be wasted by this combination (see Fig. 30).

It is necessary in all cases to ensure that there shall be no tendency for the fresh air to find its way to the extraction gratings without being thoroughly mixed with the air of the room.

Position of Extraction Gratings.

The position of the outlets from a room is determined jointly by a consideration of the position and character of the inlets, and of the possibility of their giving rise to perceptible draughts if people sit near them.

In determining these positions regard should be paid to the desirability of removing the air from positions where it is likely to be the most vitiated. It is necessary also to avoid any tendency for the air to pass direct from the inlets to the outlets without mixing with the air in the room.

Where the inlet air is warmed there is small possibility of short circuit, if the inlet and outlet are placed fairly close together, particularly where the inlet is slightly above the outlet.

In these circumstances the air will not short circuit from one grating to the other. It will rather take a circuitous path right round the room, as shown in Fig. 31, and the air removed by the extraction gratings will probably contain none of the incoming fresh air except after it has passed right round the room.

This arrangement, however, though it is probably the best practically possible for rooms of moderate sizes, up to, say, 10,000 cubic feet, as far as uniformity of distribution of fresh air is concerned, is liable to cause noticeable down draughts on the far side of the room, unless some radiation is fixed there.

In ordinary circumstances the further the outlet and inlet are apart the better will be the distribution, unless the inlet is placed at or near the floor level at one end of the room, and the outlet at the ceiling level at the other end of the room. With this arrangement, if the inlet air is warmed it will tend to rise directly to the ceiling, it will travel along the ceiling and pass away by the outlet without mixing with the main body of the air in the room at all.

Generally speaking, if the fresh air is introduced at a temperature higher than the ordinary room air the outlet should not be placed at the ceiling. The inlet should in most cases be above the outlet.

There are some great advantages to be secured by placing the main outlet at the floor level, in spite of the fact that vitiated air, owing to its temperature, tends to rise and not to fall.

It has been pointed out that the floor level is the region of the greatest dust accumulation, and therefore an outlet placed at the floor level, especially if there is no sill or any obstruction higher than the actual level of the floor, such as the frame of a grating, the outlet will carry away a great deal of the dangerous dust, especially if a vigorous draught is maintained up it. If an outlet simply consists of a hole through a wall it will almost certainly give rise to uncomfortable draughts if the wind is directed towards the opening on the outside.

Such openings may work well if the wind is in the opposite direction. In some cases an attempt is made to prevent the wind blowing through it by flap valves and similar devices. These generally give rise to a very irritating noise.

If more than one such opening is provided in the outer wall of a building, when the wind sets in certain directions, the air will blow into one of these openings and out of another. Any person sitting in between the two may come into the path of a draught of very cold air.

Even in cases where such extraction openings are provided with a propeller type of fan, which is a common practice, a fairly strong wind blowing against the outside wall will entirely overcome the suction caused by the fan and will blow air into the room in spite of the fan working at full speed.

The connection of the outlet openings by means of flues in an open trunk is treated of in another chapter.

CHAPTER IX.

CALCULATION OF QUANTITIES IN VENTILATION.

I.—Quantities of Air Required.

THE object of all ventilation is, of course, to keep the air as pure as possible. To speak of the ventilation of any room as perfect, would imply that the quality of the air in the ventilated space was at least equal to that of the outside air.

So far as medical science can pronounce, in the present state of knowledge, the ordinary outside air is generally better for purposes of health than any vitiated air after any kind of treatment yet discovered.

To limit the supply of air to a closed space occupied by human beings implies a certain degree of vitiation of the interior over that of the atmosphere. An unlimited supply of air would mean that the inner air was in free communication with the outer air.

In cold weather this would of course mean the temperature of the air in the apartment would not be very much above the outer temperature.

Any heat that could be usefully provided in the apartment could only be radiated heat, as heat given off by conduction to the air from such devices as "radiators" would immediately be wholly lost.

The expense of maintaining any perceptible degree of heat in the air of a room, ventilated on the open air principle, would be enormous. It would not, in fact, be possible as a practical proposition to maintain the air temperature many degrees above the outside temperature. The presence of an open fire or other powerful radiant in the room would, however, give a *feeling* of warmth, which might be sufficient for the comfort of occupants of a room who were accustomed to that kind of ventilation, although the actual temperature as indicated by a shielded thermometer might be the same as in the outer air.

At any rate, in the present state of public opinion, such ventilation as this is an impossibility, and need not be further considered. It appears probable, however, that one or two generations hence the advantages of this kind of ventilation may be so far appreciated that it will become common. Until that time arrives, however, the heating engineer has to deal with things as he finds them.

The relation between the quantity of air introduced into a room per hour and the state of the air in it, is a mathematical relation which can be easily calculated. The desirable or necessary condition of the air in the room is a medical or hygienic matter. The engineer is only concerned in obtaining the results which are dictated to him by medical and hygienic authorities, and it is to that authority that he looks for specification of the requirements.

The calculation respecting the ventilation of a room consists of three parts—

(1) The quantity of air required to be impelled through the room per hour.

(2) The power required to move this quantity of air, including that required to overcome the resistance of the flues, if any.

(3) The quantity of heat required for warming, and of water for humidifying the air, together with auxiliary calculations respecting the means for cleaning the air, and other similar matters, all of which will be separately dealt with.

Quantity of Air Required.

The estimation of the quantity of air required to be delivered to a room per hour varies according to the purposes for which the air is required. So far as present knowledge goes, the only calculable functions which the introduction of fresh air into a room subserves are—

(1) To ensure that the percentage composition of CO_2 of the room air shall not exceed a certain maximum.

(2) That the temperature in the room shall not exceed a certain maximum.

(3) To carry heat to the room in cases where the air is used as a circulating medium.

(4) To carry off moisture from the room or its contents.

1. It will be seen from the chapter on the hygiene of ventilation to what extent vitiation of the air of a room is caused by the presence therein of human beings and of gas lights.

In order that a calculation may be made of the amount of air interchange required, it is necessary to have specified what is the highest allowable percentage of CO_2 in the inner air, under certain precisely defined conditions.

The general composition of the outer air may be taken as being 4 parts of CO_2 per 10,000 of air, and for ordinary purposes 10 parts of CO_2

per 10,000, or 1 part in 1,000 in the inner air must not be exceeded. This rate of ventilation is not to be considered good.

Six parts per 10,000 is considered extremely good; 7 parts per 10,000 good; 8 parts, medium; 9 parts, moderately bad; 10 parts, the lowest allowable; though up to 15, and even more, parts are countenanced in certain circumstances by eminent authorities, especially in cases where the CO_2 has a non-animal origin.

Assume that the average amount of CO_2 given off by a human being is constant and = 0.70 cubic feet per hour. Then it is clear that every degree of purity of the room air must correspond to a definite quantity of air to be introduced into the room per head per hour.

The general investigation of the state of the room air at any time is as follows :—

Let V be the volume of fresh air taken into the room in cubic feet per hour.
C The contents of the room in cubic feet.
a The proportion of CO_2 in one cubic foot of pure atmospheric air.
k_1 The proportion of CO_2 in one cubic foot of air in the room at the beginning of the ventilation, and
k_2 At the end, after
T Hours.
n The number of persons in the room, and each producing
b Cubic feet of CO_2 per head per hour.

Then if the proportion of CO_2 in parts of air at any moment t hours after the commencement of the ventilation is indicated by k, the total quantity of CO_2 in the room at that moment is

$$C \times k \text{ cubic feet.}$$

Total quantity of CO_2 in room at beginning $= C \times k_1$.
,, ,, ,, end $= C \times k_2$.

Consider a short interval of $d\,t$ hours just after the instant in question, during which the *proportion* of CO_2 increases by the amount $d\,k$.

The increase in the total amount of CO_2 added to the room during this short interval is

$$C \times d\,k \text{ cubic feet.}$$

But this increase must have been due to the excess of the amount of CO_2 added to the room air during the interval over that removed by the ventilation.

The CO_2 added to the room air comes from two sources—

(1) That coming in with the fresh air.
(2) That given off by the n human beings in the room, each of whom produces b cubic feet per hour.

The CO_2 removed from the room is simply that contained in an average sample of the air removed by ventilation, which we must take as the same quality as the average of the room air.

The magnitude of these various quantities is as follows :—

(1) $V\ a\ dt.$
(2) $nb\ dt.$
(3) $V\ k\ dt.$

So we have the equation

$$C\,dk = V\,a\,dt + nb\,dt - V\,k\,dt = dt\,(Va + nb - V\,k),$$

or

$$\frac{dt}{C} = \frac{-\,dk}{V\,k - V\,a - nb}$$

Integrating this we have

$$\frac{t}{C} = \frac{1}{V} \times \log_e (V\,k - V\,a - n\,b) + \text{const.}$$

Taken between the limits $t = o$ and $t = T$, for which value $k = k_1$ and $k = k_2$ respectively.

$$\frac{T}{C} = \frac{1}{V} \log_e \frac{V\,k_1 - Va - nb}{V\,k_2 - Va - nb}$$

or

$$T = \frac{C}{V} \times \log_e \frac{V\,k_1 - Va - nb}{V\,k_2 - Va - nb}$$

This result enables the time necessary for the air to be deteriorated from one condition to another to be calculated.

Thus, assume in a hall at the commencement of a performance the proportion of CO_2 is 6 per 10,000.

Number of people present, 1,000.
Content of hall, 260,000 cubic feet.
Fresh air, three changes per hour.
Required the time until the CO_2 becomes 10 parts in 10,000.

In this case we have

C	260,000
k_1	.0006
k_2	.001
V	780,000
a	.0004
$n =$	1,000
$b =$	0.7

Hence

$$T = \frac{260{,}000}{780{,}000} \times \log_e \frac{780{,}000 \times .0006 - 780{,}000 \times .0004 - 1{,}000 \times 0.70}{780{,}000 \times .001 - 780{,}000 \times .0004 - 1{,}000 \times 0.70}$$

$$= \tfrac{1}{3} \log_e \frac{468 - 312 - 700}{780 - 312 - 700} = \tfrac{1}{3} \log_e \frac{544}{232}$$

$$= \tfrac{1}{3} \log_e 2.35 = \tfrac{1}{3} \times .8544$$

$$= .285 \text{ hours} = \text{about } 17 \text{ minutes.}$$

When the carbonic acid has attained its maximum in any given conditions, so that there is a uniform percentage of CO_2 in the room air, it is clear that the amount of CO_2 carried away by the extract air must be exactly equal to the amount received jointly from the carbonic acid in the inlet air and that received from the breath of the occupants of the room. In this case

$$V k_2 = Va + nb$$

or

$$V = \frac{Va + nb}{K_2} \text{ or } K_2 = \frac{Va + nb}{V} \text{ or } K_2 = a + \frac{nb}{V}$$

In such circumstances, of course, the right hand term in the logarithmic equation above becomes infinite, and the final conditions denoted by k_2 would only be attained in the limit when t was infinitely great.

The diagram given in Fig. 33 is calculated from this latter equation, and shows the quantity of air which must be supplied to a room per head per hour in order to maintain different standards of purity.

The standard of purity to be specified is a matter for the hygienic and medical authorities.

The above logarithmic equation is only applicable during the time when the composition of the air is appreciably deteriorating. It can also be used for ascertaining the amount of ventilation which has actually been proceeding in a room when the condition at the beginning and end of a certain interval is known. For instance, if it is known that after the above room has been occupied for some time the carbonic impurity is, say, 24 parts per 10,000. If, after the occupants have left the room, observations are kept of the state of the air at subsequent intervals, and it is found that in, say, 35 minutes the air has improved, so that it is then 13 parts per 10,000.

The application of this equation will show what is the corresponding value of V, for in this case $nb = 0$, and the equation becomes

$$V = \frac{C}{T} \log_e (V k - V a)$$

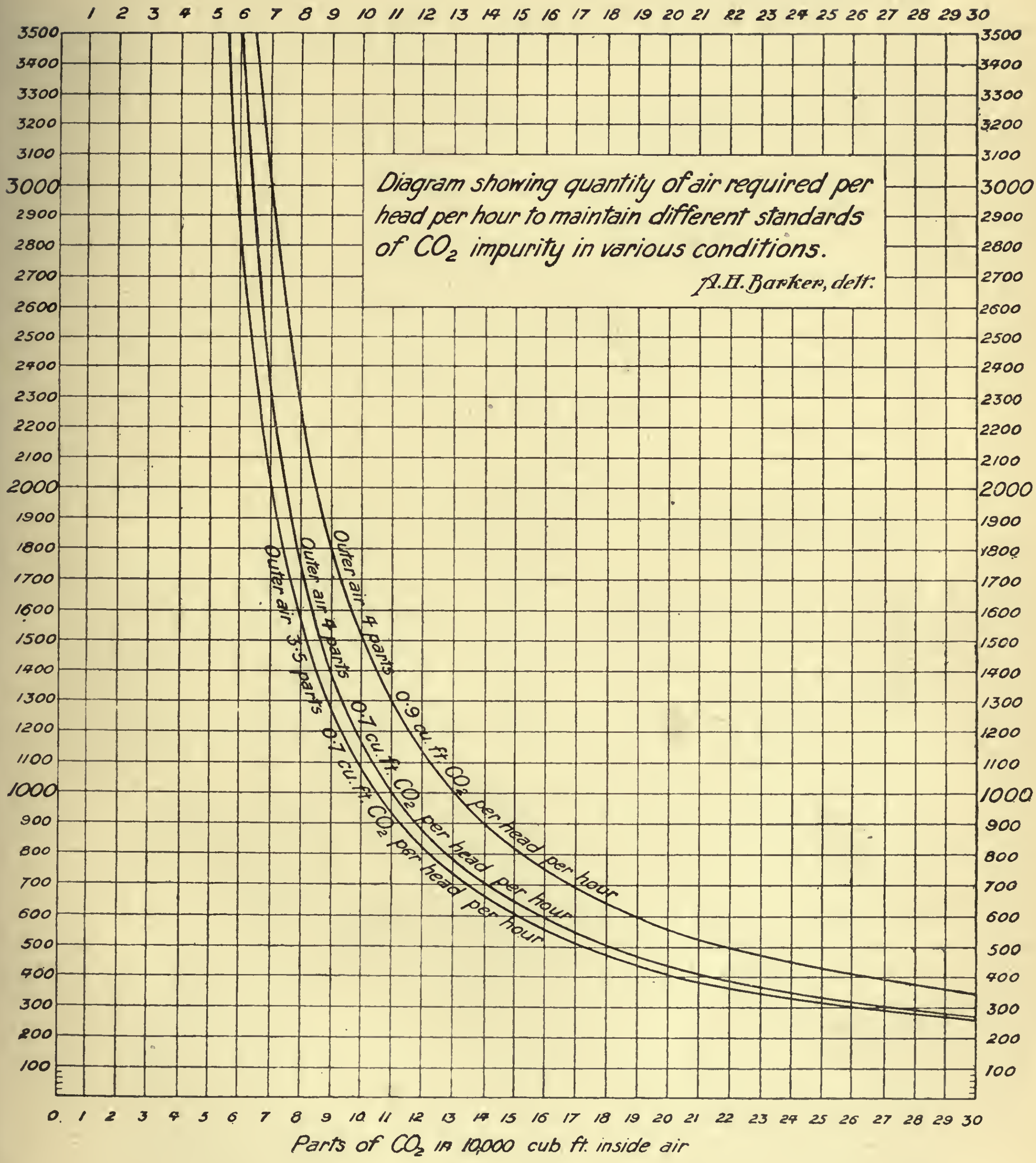


Diagram showing quantity of air required per head per hour to maintain different standards of CO_2 impurity in various conditions.
A.H. Barker, delt.
Outer air 4 parts 0.9 cu. ft. CO_2 per head per hour
Outer air 4 parts 0.7 cu. ft. CO_2 per head per hour
Outer air 3.5 parts 0.7 cu. ft. CO_2 per head per hour
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30
3500 3400 3300 3200 3100 3000 2900 2800 2700 2600 2500 2400 2300 2200 2100 2000 1900 1800 1700 1600 1500 1400 1300 1200 1100 1000 900 800 700 600 500 400 300 200 100
0. 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30
Parts of CO_2 in 10,000 cub ft. inside air

FIG. 33.

or taken between limits

$$V = \frac{C}{T} \log \left(\frac{k_1 - a}{k_2 - a} \right)$$

Here

C = as before, 260,000.
T = 35 minutes = 0.585 hours.
k_1 = .0024.
k_2 = .0012.
a = .0004.

$$V = \frac{260{,}000}{.585} \log_e \frac{.0024 - .0004}{.0013 - .0004}$$

$$= \frac{260{,}000}{.585} \times \log_e 2.22$$

$$= \frac{260{,}000 \times .7975}{.585}$$

= 354,000 cubic feet per hour.

This is a most useful method, in fact the only accurate method of determining what is the natural interchange of air in a room.

The majority of rooms for which close calculations as to ventilation have to be made are rooms which are not continuously occupied for a long period of time. Thus, school rooms are rarely occupied more than two hours at a time, and the necessary interchange for such rooms must be calculated on the basis that at the end of two hours' occupancy the carbonic acid impurity must not exceed a certain percentage.

If the rooms are occupied longer than two hours the percentage will of course be greater.

The same is true for such buildings as churches, concert halls, etc., which are generally occupied at the outside for two hours at a time.

The equation is not applicable in cases like hospital wards, which are continuously occupied.

In order thoroughly to understand this most important matter it will be well to trace the curve represented by the equation

$$\frac{dt}{C} = \frac{-dk}{Vk - Va - nb}$$

or its integral

$$\frac{t}{C} = \frac{1}{V} \log_e (V K - V a - n b) + \text{constant},$$

so as to illustrate the way in which the air of a room is vitiated.

It will be seen by a careful examination of the preceding equation that the degree of vitiation in an occupied room only gradually attains its maximum value. At the moment when the occupants come into a room which has been in free communication with the outside air, the air in the room has the same composition as the outer air, that is to say, it is not vitiated at all.

Consider first how the air would be vitiated if there were no ventilation whatever. In this case the whole of the carbon-di-oxide given off would be retained in the room. At the commencement of the occupancy the degree of vitiation is .0004 (= a) of the total volume. Then every cubic foot of CO_2 generated in the room increases that value by $\frac{1}{C}$ where C is the contents. As the amount of CO_2 introduced is proportional to the time, the curve representing the progressive vitiation will be a straight line. Since this straight line passes through the point (0 a) and has an inclination equal to $\tan^{-1}\frac{nb}{C}$ its equation is

$$K = \frac{nb}{C}t + a$$

The diagram, Fig. 34, is drawn to scale, the particulars being taken from those given for the meeting hall on page 161, but the air being assumed unvitiated at the commencement of occupancy.

Assume next that there is a constant and regular interchange of given magnitude, and that the incoming fresh air is uniformly distributed so that the composition of the air all over the room is uniform (which in ordinary circumstances it will be almost, but not perfectly). The extract air which is driven out by the incoming fresh air will at first be pure and unvitiated. As therefore no excess of CO_2 is at first leaving the room at all, the rate of increase of vitiation for the first few minutes will be the same as though no ventilation whatever existed. Gradually, however, some of the CO_2 given off from the occupants begins to reach the extraction gratings in increasing quantity.

When this happens, the curve representing the actual vitiation begins to leave the side of the straight line and slopes over, so as to indicate a smaller actual vitiation than if there had been no ventilation. The curve keeps on sloping further and further away from the straight line until as the concentration of CO_2 in the extract air becomes more and more nearly equal to its ultimate value the concentration in the room air ceases to rise at all. When this state is reached the amount of CO_2 leaving the hall by the extraction gratings becomes equal to the amount received jointly from the breath and from the outer air.

It will be seen from this explanation that the curve of vitiation is

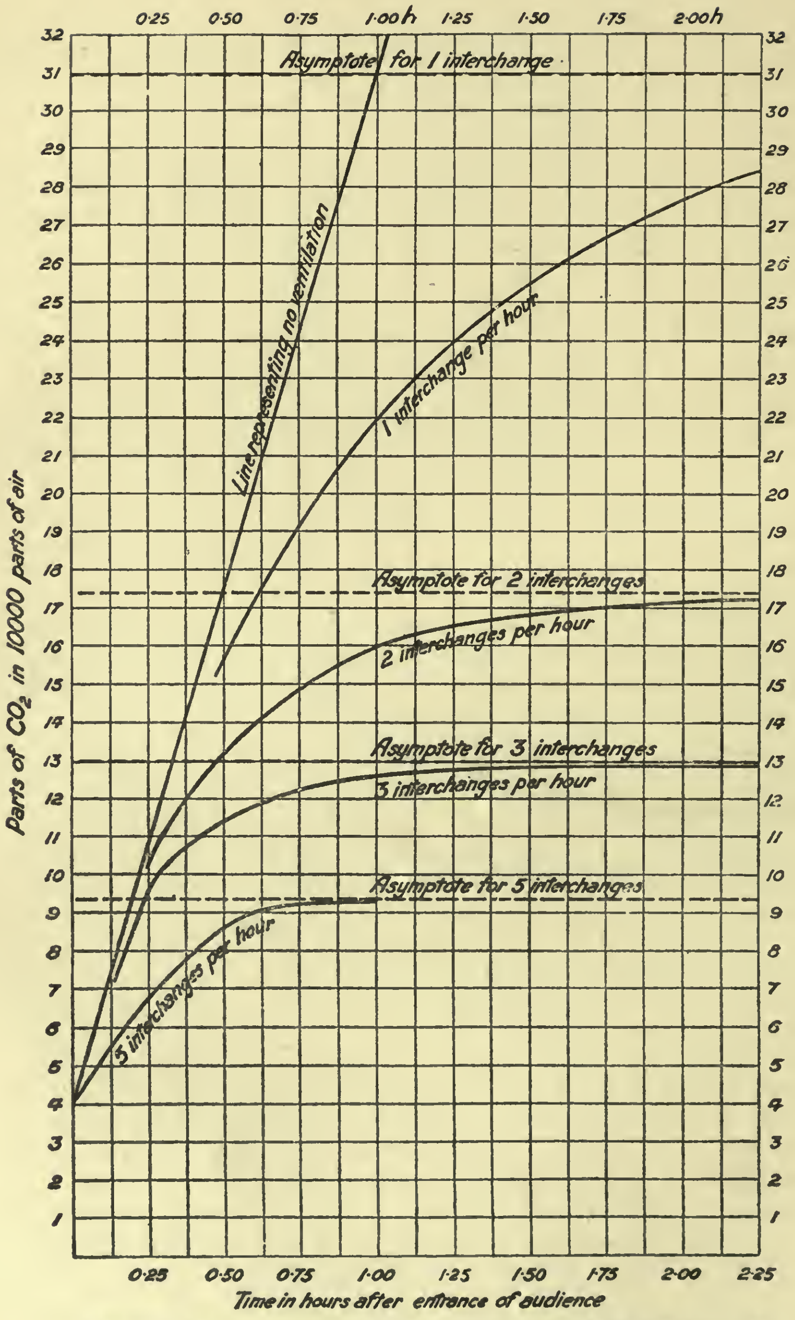
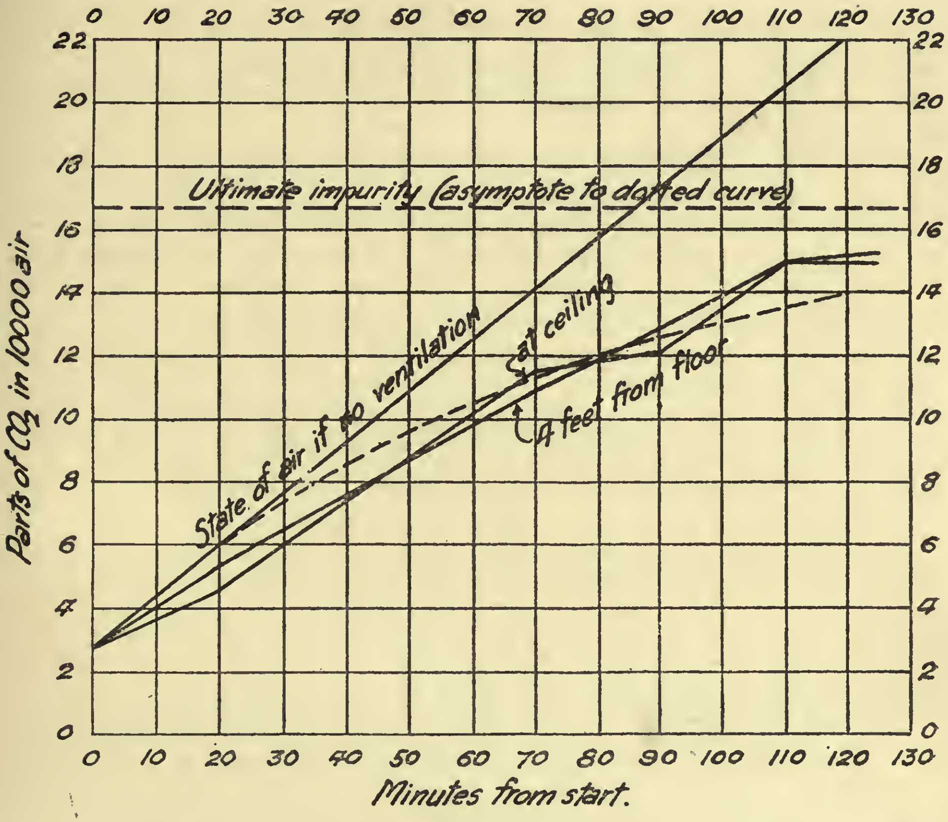


Diagram shewing progressive vitiation of air of hall of 260000 cub. feet with 1000 persons present and various interchanges of air.

A.H. Barker, delt.

FIG. 34.



Vitiation of room air illustrating comparison of results from calculation (dotted) with those derived from actual analysis of air (strong) by Haldane. Contents of room 3070 cubic feet, all openings closed. Occupants, three women and one man. Probable CO_2 3·4 cubic feet per hour, probable fresh air 2470 cubic feet per hour.

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FIG. 35.

initially tangential to the no ventilation curve, and also is asymptotic to the horizontal line

$$K = a + \frac{nb}{V}$$

representing the ultimate degree of concentration, that is to say, the curve approaches the asymptote nearer and nearer the longer the time t is taken, but never quite reaches it, if the above specified conditions are precisely fulfilled.

Of course, in practice such precisely defined conditions are never fulfilled. There is always a variation from hour to hour or from minute to minute in the amount of air finding its way into the room. Such changes being brought about by slight changes in the external wind velocity of the opening of doors, etc. The curve, therefore, will only give an approximate idea of what the actual ventilation will be.

After a certain time, for practical purposes, the curve is indistinguishable from the asymptote.

The smaller the amount of ventilation the longer it takes for the curve to attain its horizontal direction. On the diagram five curves have been plotted out, representing 5, 3, 2, 1, and 0 interchanges per hour respectively.

The alteration in the shape of the curve is most instructive and should be carefully studied. All these curves are plotted from the formula

$$\frac{t}{C} = \frac{1}{V} \log_e \left(\frac{Vk_1 - Va - nb}{Vk_2 - Va - nb} \right)$$

corresponding values having been taken in each case for V.

In order to compare the theoretical with the actual state of the air, the curve has been traced in Fig. 35 for a particular case, in which exact experiments have been made by a competent authority. Haldane reports a test on a room having a cubic capacity of 3,070 cubic feet and 11 feet high, when occupied by three women and one man. The day was bright and sunny and there was no perceptible wind. The house was in the country and all the openings were closed.

The result of that experiment is shown in strong lines on the accompanying diagram, Fig. 35. It may be assumed that the production of CO_2 from the four persons in the room was about 3.4 cubic feet per hour, and the inlet of fresh air may be therefore calculated as being about 2,440 cubic feet per hour.

Given these conditions, the dotted curve shows the value of the theoretical impurity throughout the experiment, while the firm lines show what was actually observed by careful analysis. A careful study of this curve, in conjunction with the formula, is strongly recommended.

The equation will also reveal some straightforward and useful methods of determining rapidly what will be the degree to which the impurity will finally rise when the actual impurity after a certain interval is known.

If, as before, a denotes the proportion of CO_2 in the outer air, and k the variable proportion of CO_2 in the inner air, then $(k - a)$ denotes the excess of impurity in the inner air over the outer air. Let this be denoted by E.

Now if the air in the room were not changed at all, that is, if the room were perfectly air-tight, the amount of CO_2 in the air after n persons had occupied the room for t hours would be $n\ b\ t$ cubic feet in excess of the number of cubic feet in an equal volume of fresh air.

The proportionate excess which this addition of CO_2 would represent would be

$$\frac{nbt}{c}$$

Let this be denoted by E.

If V cubic feet of air comes into the room per hour, then the time required for completely changing the air of the room once would be $\frac{C}{V}$ hours. Let this be denoted by T. Then

$$\frac{E}{E_0} = \frac{K - a}{\frac{nbt}{c}} = \frac{(K - a)\,c}{nbt}.$$

$$E_0 \times \frac{T}{t} = \frac{nbt}{c} + \frac{T}{t} = \frac{nbT}{c} = \frac{nb}{V} = \text{ultimate excess.}$$

It will be seen from the equation that for every value which the fraction $\frac{E}{E_0}$ assumes, there is a constant value for $\frac{T}{t}$ so that if the value of E is determined by experiment, the time t being known, the value which E will ultimately attain can be found by multiplying E_0 by the corresponding value of $\frac{T}{t}$.

The table in the Appendix showing the corresponding values of $\frac{E}{E_0}$ and $\frac{T}{t}$ is due to Mr. P. J. Kirkby.

For instance, consider the case of the hall described on page 161. Assume that an analysis of the air is made after the audience have been sitting for three-quarters of an hour, and it reveals a total impurity of 12 parts of CO_2 per 10,000 of air, the outside air being 3.5, and the occupants being assumed to give off 0.9 cubic feet of CO_2 per hour. Here we have

$$E = 12 - 3.5 = 8.5 \text{ parts per } 10{,}000.$$

Also

$$E_0 = \frac{nbt}{c} = \frac{900 \times 0.75 \times 10{,}000}{26{,}000} = 26 \text{ parts per } 10{,}000.$$

$$\frac{E}{E_0} = \frac{8.5}{26} = 0.327.$$

The table shows that the corresponding value of $\frac{T}{t}$ is .34, or time of one interchange = 0.34 × 0.75 = 0.255.

This gives the rate of interchange

$$\frac{260{,}000}{0.255} = 1{,}020{,}000 \text{ cubic feet per hour.}$$

Hence the value of E_1 is 0.34 × 26 = 8.8 parts above the outer air. The final degree of impurity is therefore

$$8.8 + 3.5 = 12.3.$$

For rough general purposes the following short table will serve :—

Where the value of $\frac{E_0}{E}$ is ...	3 or more	2	1.75	1.5	1.25
The probable future maximum excess will be E × ...	1	1.24	1.4	1.7	2.7

2. It is sometimes required to ascertain the amount of air, at a given outside temperature, which it is necessary to introduce in order to prevent the temperature in the room from rising above a certain limit under certain conditions. This requirement generally arises in cases, somewhat rare in this climate, in which provision is to be made against too high a summer temperature, in such cases as theatres, churches, dance rooms, and so forth. The same investigation also serves for cases where the whole or part of the heat supplied to a room is derived from the heat communicated to the air.

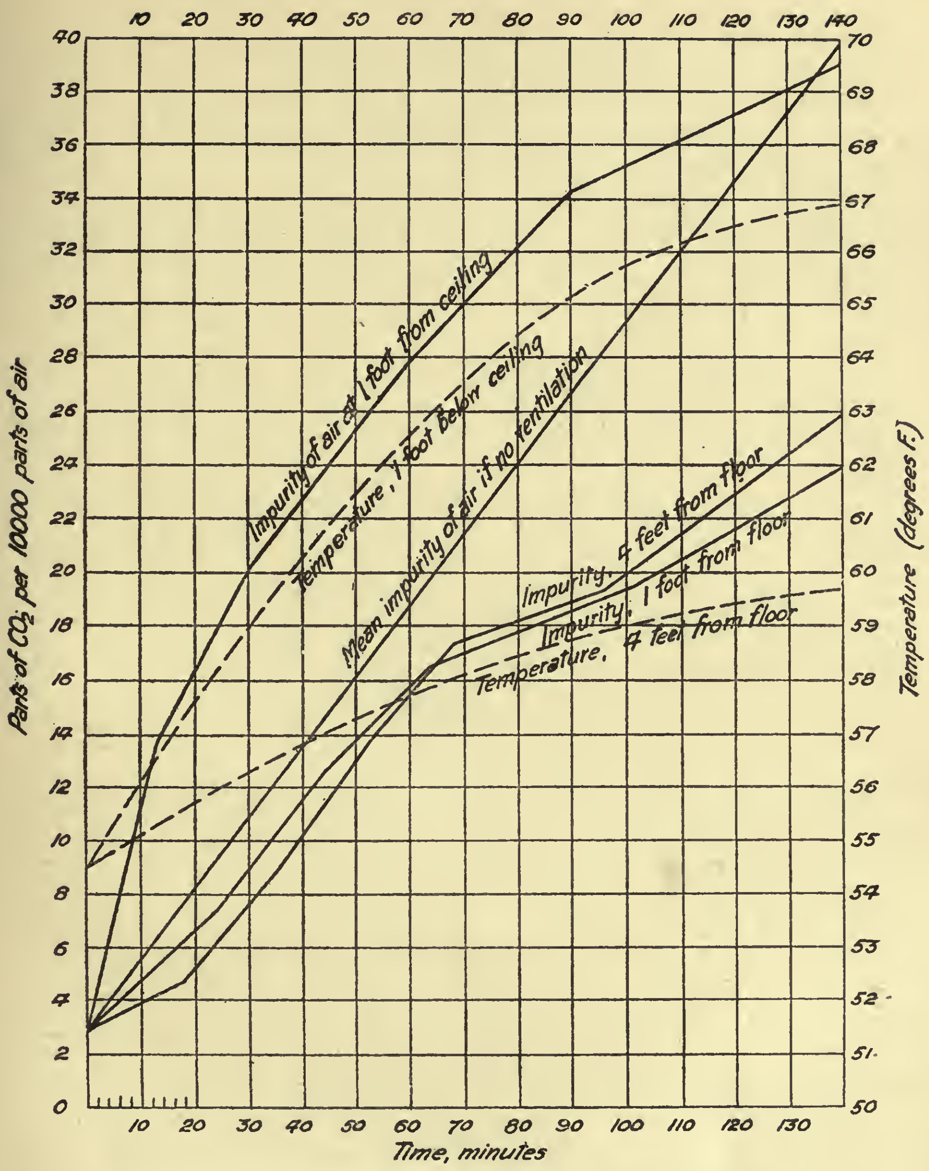
Assume that a number of persons are assembled in a hall, whose cubic content is C cubic feet, and the outside temperature of the air is t_0, and the temperature which is not to be exceeded is t_1. Let

W be the quantity of heat necessary to be removed.

W_1 the quantity of heat given off by the n people present, which may be taken as $n\ h$ where h is the heat given off by one person, usually 400 B.T.U. per hour.

W_2 the amount of heat derived from any other source, such as gas lighting, auxiliary radiators, fires, etc.

W_3 the quantity of heat which is either passed into or out of the room by conduction from the walls, etc.



Experimental curves illustrating rise of temperature (dotted) and progressive vitiation of air (strong) by one person and three gas jets 6½ feet above floor burning jointly 15 cubic feet of gas per hour.
Contents of room 5700 cubic feet, all openings closed.
Total probable production of CO_2, 9·1 cubic feet per hour

A.H.Barker, delt.

FIG. 36.

Then obviously

$$W = W_1 + W_2 \pm W_3.$$

The heat will be carried off by means of air having a temperature t_i, that is the assumption is made that the mean temperature of the room is the same as that of the extract air.

Let x be the quantity of air required in cubic feet per hour, measured at the internal temperature t_i. Then the weight of this quantity of air is

$$\frac{x}{1 + a\,t_i} \times .0863 \text{ (see page 31)}$$

where $a = \frac{1}{461}$, and t_i = temperature Fahrenheit.

In raising this weight of air from a temperature t_o to t_i it must therefore absorb

$$\frac{x}{1 + a\,t_i} \times 0.0863 \times 0.238 \times (t_i - t_o) \text{ B.T.U.}$$

whence

$$x = \frac{(W_1 + W_2 - W_3)(1 + a\,t_i)}{0.0206 \times (t_i - t_o)}$$

or since

$$W \text{ B.T.U.} = W_1 + W_2 \pm W_3$$

$$x = \frac{W(1 + a\,t_i)}{0.0206\,(t_i - t_o)}$$

The quantity of heat (W_1) given off by the people present may be ascertained from the previous chapter, from which it will be seen that an adult human being at rest gives off about 400 B.T.U. per hour, so that the total heat given off by n persons = 400 n B.T.U. per hour.

The other quantities can also be ascertained, either from that chapter or from the chapters on the loss of heat from buildings and radiators.

Example.—Required the volume of air necessary to be introduced per hour into a hall of 260,000 cubic feet, with an outside temperature of 50° F., in order to prevent the temperature from rising above 60° when 1,000 people are present, assuming the heat lost from the walls and windows, etc., per degree difference is 20,000 B.T.U., and assuming that 150 cubic feet of gas are burnt per hour in lighting the room, whose calorific value is 550 B.T.U. per cubic foot.

Here

$$W_1 = 400{,}000$$
$$W_2 = 150 \times 550 = 82{,}500$$
$$W_3 = 10 \times 20{,}000 = 200{,}000$$
$$t_i = 60$$
$$t_o = 50$$

$$x = \frac{(482{,}500 - 200{,}000)\left(1 + \frac{1}{461}60\right)}{.0206 \times 10} = \frac{282{,}500 \times 1.13}{.206}$$

$$= 1{,}540{,}000 \text{ cubic feet per hour.}$$

In the same way as above the time required to raise any required temperature could be calculated, but this time would obviously vary according to the position of the extraction grating. If the extraction grating were at the floor level the air that it would take away would in fact be much cooler than if it were at the ceiling level, and therefore more heat would be retained.

Any calculations, however, based on these assumptions cannot be expected to agree very closely with the practical results on account of the leakiness of the walls. The matter, therefore, cannot be profitably gone into more in detail.

3. Consider the case of a room whose temperature is to be maintained by an inflowing current of warm air. The temperature of the air flowing in is not to exceed t_1°.

Assume that the loss of heat from the walls is calculated as W_3 B.T.U. and the amount of heat derived from radiators and gas lights is calculated as W_2 B.T.U., W_1 being, as before, the quantity of heat given off by the persons present, namely,

$$400 \times n$$

where n is the number of people present.

Let V be the number of cubic feet of air introduced, reckoned at the temperature t_i. Its weight is

$$\frac{V}{1 + a\, t_i} \times .0863$$

and the heat it contains reckoned above zero Fahrenheit

$$\frac{V}{1 + a\, t_i} \times .0863 \times 0.238 \times t_1 \quad \text{(see page 38).}$$

The heat lost in the same weight of extraction air leaving the hall at t_i is

$$\frac{V}{1 + a\, t_i} \times .0863 \times 0.238 \times t_i$$

whence the balance of heat available for maintaining the temperature is

$$\frac{V\,(t_1 - t_i) \times .0206}{1 + a\, t_i} = W_3 - W_1 - W_2$$

whence

$$V = \frac{(W_3 - W_1 - W_2)(1 + a\, t_i)}{(t_1 - t_i)\, .0206}$$

EXAMPLE.—Apply this formula to the same large hall as before, and assume the outside temperature is 32° F., the desired inside temperature is 60° and the maximum allowable inlet temperature is 70° F., then

$$V = \frac{(28 \times 20{,}000 - 400{,}000 - 82{,}500)\left(1 + \frac{70}{461}\right)}{.0206 \times 10}$$

$$= \frac{77.500 \times 1.157}{.0206}$$

$$= 435{,}000$$

4.—Calculations have sometimes to be made in order to ascertain the amount of air which it is necessary to supply in order to dry a certain weight of clothes, or wood, or other material per hour.

An exact calculation for this purpose can only be made when the water to be removed is on the surface of the material. For in cases like the drying of wood the moisture to be removed is buried deep in the wood, and the time required to dry it depends more on the properties of wood than on the quantity of air passed through.

In the case of clothes drying, for such purposes as laundry drying rooms, the calculation is as follows :—

Let it be required to dry w pounds of wet clothes which contain as they come from a centrifugal wringer, say, 75% of their weight in water, and let it be required to dry the clothes completely in T hours. The following is a theoretical calculation, but it must be mentioned that the perfection and the rapidity with which the garments will be dried in practice depends on the distribution of the air in the chamber.

If, as is sometimes done, the air is all introduced at one point, the extraction air all taken out at another point in the chamber, there is no certainty that the air will be equally distributed, and therefore those clothes in the path of the main air currents will be dried rapidly, while the others may remain wet for a long time.

The moisture in the air leaving a drying-room is generally about from 85% to 95% of complete saturation. Assume this fraction is 90%.

Let V be the volume of air forced into the chamber per hour, measured at the cold inlet temperature t_1.

Let the amount of moisture in 1,000 cubic feet of this inlet air be m_1 pounds, and the temperature $t_1°$.

Let the temperature of the air on leaving the chamber be t_2.

Let ms_2 be the amount of moisture in pounds in 1,000 cubic feet of saturated air at temperature t_2.

Then the amount of moisture which would be contained in 1,000 cubic feet of the air after being heated to t_2, but before extracting any moisture from the clothes, would be

$$m_1 \times \frac{1 + a\,t_1}{1 + a\,t_2} \text{ pounds (see page 31).}$$

After passing through the chamber the volume of what was originally 1,000 cubic feet of cold air measured at t_1 will be

$$1{,}000 \times \frac{1 + a\,t_2}{1 + a\,t_1} \text{ cubic feet of air at } t_2.$$

The total amount of moisture actually in it at, say, 90% of saturation will be

$$\frac{9}{10} \times m_{s2} \times \frac{1 + a\,t_2}{1 + a\,t_1} \text{ pounds,}$$

of which the amount of moisture originally in the air was m_1 lbs.

Thus, the amount of moisture carried off from the clothes by each thousand cubic feet of air, measured at the temperature t_1,

$$\frac{9}{10} \times m_{s2} \times \frac{1 + a\,t_2}{1 + a\,t_1} - m_1$$

Now the total quantity of moisture to be carried off is

$$\frac{W \times \frac{75}{100}}{T} \text{ pounds per hour,}$$

of which amount each 1,000 cubic feet, measured at the original temperature, carries off a quantity as above. Therefore the required number of thousands of cubic feet per hour is

$$\frac{\dfrac{W \times \frac{75}{100}}{T}}{\dfrac{9}{10} \times m_{s2} \times \dfrac{1 + a\,t_2}{1 + a\,t_1} - m_1}$$

It must be mentioned, however, that in practice the 90% degree of saturation of the outlet air would not be maintained until the drying was completed, but that the increase in the time required would be almost compensated for by the rise in temperature of the air as the drying proceeds.

The first effect of the introduction of wet clothes into a drying chamber through which a strong current of hot air is passing is to reduce the temperature 25° or 30°, or even more, according to circumstances. The general initial temperatures employed for linen drying are

Linen and cotton goods	120° to 140° F.
Blankets and flannels	90° to 110° F.

The lower the temperature at which the drying is effected and the greater the volume of air employed the better is the result. The higher its temperature the greater is the risk of shrinkage and the greater the hardening effect on the material.

To express the above calculation in the form of a general equation would be unnecessarily complicated, seeing that it would involve the somewhat complicated expression relating to the evaporation of moisture.

EXAMPLE.—It will be seen from the chapter on the Hygiene of Ventilation that it is the opinion of many authorities on hygiene that the amount of moisture in air is a better criterion of the quality of the air for breathing purposes than is the proportion of CO_2. It is therefore quite possible or probable that in future the amount of moisture in breathing air will be one determining criterion of the perfection of ventilation. If this is so it will be necessary to make calculations to determine how much air is to be introduced into a room in order that the relative humidity may be kept below certain limits. The following is a method to be adopted in this case :—

Consider the same hall as discussed in the previous examples, having 260,000 cubic feet, there being 1,000 persons present. Assume that the degree of moisture in the outside air is 78% and the temperature 38° F. Determine the amount of air at this temperature and of this degree of moisture which must be introduced into the hall in order that the relative humidity may not at any time exceed 70% with an internal temperature of 65°. Here the moisture given off from the bodies of the persons present is $1,000 \times 0.15$ pounds of water per hour.

Let it be assumed that x cubic feet of air have been introduced into the room per hour. It will be seen from the table that every thousand cubic feet of entering air introduces into the room an amount of moisture equal to about 0.38 pounds.

$$x \text{ cubic feet therefore contain } \frac{x}{1,000} \times 0.38 \text{ lbs. of moisture.}$$

As x represents the volume of air introduced, measured at 38°, after being heated up to 65°, the volume will become

$$x \times \frac{461 + 65}{461 + 38} = 1.057\,x$$

If the moisture in this air is at 70% saturation, the amount of water it contains per thousand cubic feet is 0.97 pounds. The result of introducing into the hall x cubic feet of air per hour, measured at 38°, is, of course, to displace the quantity 1.057 x cubic feet of air at 65°. This carries away an amount of moisture equal to $\frac{1.057\,x \times 0.97}{1{,}000}$ pounds per hour.

It is evident, therefore, that the amount of moisture introduced into the room with the fresh air + the amount of moisture given off by the persons present must in a state of equilibrium be equal to the amount of moisture carried away by the foul air. Hence we have the equation, allowing vapour emitted per man per hour = 0.15 lbs.

$$1{,}000 \times 0.15 + \frac{x}{1{,}000} \times 0.38 = \frac{1.057 \times 0.97 \times x}{1{,}000}$$

from which the value of x can be calculated as follows:—

$$150{,}000 = 1.022\,x - 0.38\,x$$

$$x = \frac{150{,}000}{0.642} = 233{,}000 \text{ cubic feet.}$$

If it is required to ensure that the amount of moisture after, say, a period of 1½ hours shall not exceed 70% humidity, the procedure would be somewhat different. The general method is similar to that shown on page 160. It can easily be adapted for the purpose of calculating the time rate of rise of humidity.

CHAPTER X.

CALCULATIONS OF QUANTITIES IN VENTILATION.

II.—Development of Pressure Necessary for Moving the Requisite Quantity of Air.

We have now to consider the means employed for developing the power required to move the quantity of air.

"Natural Ventilation."

Consider the state of static pressure of the air in a room which is at a different temperature from the outer air.

Assume that the room is entirely enclosed by a wall, roof and floor, which are all air-tight except for a small hole at the floor level, communicating with the outer air, as shown in Fig. 37. The room is H feet high. The outside temperature is t_0 and the inside temperature t_1, and is uniform all over the room. The latter is a condition of things rarely found in practice, though for a room of moderate height (up to, say, 10 feet) the variation from uniformity is not great if the heating is properly done.

When t_0 and t_1 are so nearly equal that there is no appreciable difference in density between the inner and outer air, the pressure at any level within the room will be the same as at the same level outside. When, however, the temperature in the room is different from that of the outer air, this is not the case.

Suppose the absolute pressure of the external air in pounds per square foot at the level of the floor of the room is p_f. The difference in absolute pressure between the floor level and the ceiling level will be insignificant when compared to the absolute pressure, but cannot be disregarded when differences of relative pressure are in question. The absolute amount of the difference in pressure between floor and ceiling levels will be the weight of H cubic feet of air per square foot. If the height H is taken as 12 feet, the magnitude of this will be 12 × 0.08 or about, say, 0.96 lb. per square foot = 0.185 inches of water.

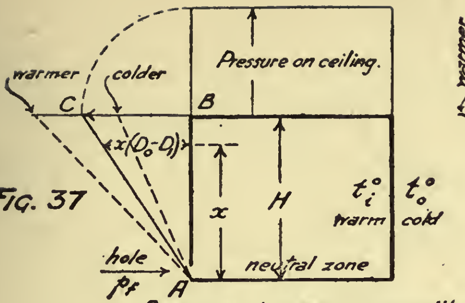


FIG. 37

Pressures in warm room with one opening at floor level.

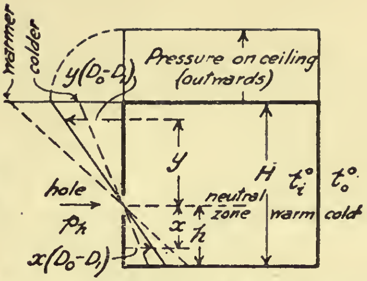
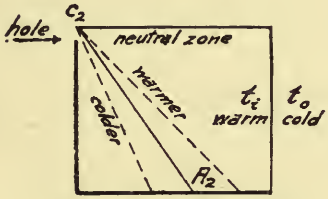


FIG. 38

Pressures in warm room with hole in middle of wall.

FIG. 39



Pressures in warm room with hole at ceiling level.

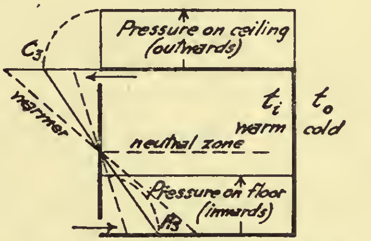
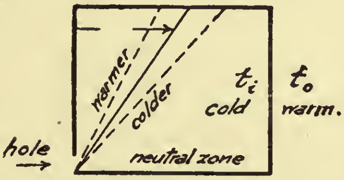


FIG. 40

Pressures in warm room with openings at floor and ceiling. Ventilation proceeding.

FIG. 41



Pressures in cold room with hole at floor.

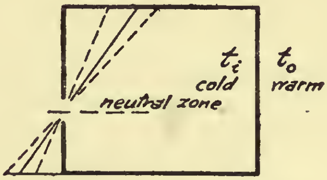
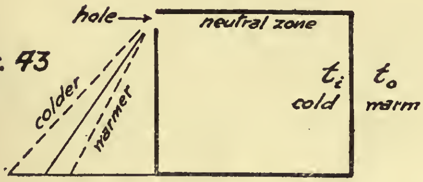


FIG. 42

Pressures in cold room with hole in wall.

FIG. 43



Pressures in cold room with hole at ceiling

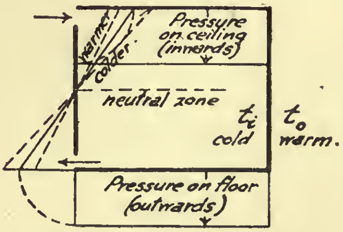


FIG. 44

Pressures in cold room with holes at ceiling and floor. Ventilation proceeding.

Suppose the density of the outer air at the temperature t_o° is D_o, and at the inner temperature t_i° is D_i. Then at a distance x feet above the floor of the room, the outside pressure will be nearly

$$p_f - x\ D_o$$

while the pressure at the same level inside the room will be nearly

$$p_f - x\ D_i$$

Now as D_i is less than D_o if the room air is warmer than the outside air, it follows that the pressure inside will be greater than the pressure outside at any level above the floor by the amount

$$(p_f - x\ D_i) - (p_f - x\ D_o) = x\ (D_o - D_i)$$

If the room has the height H, the absolute pressure inside at the ceiling will be

$$p_f - H\ D_i$$

and the pressure outside at the same level will be

$$p_f - H\ D_o$$

The difference in pressure, therefore, at the ceiling will be

$$H\ (D_o - D_i)$$

The difference of pressures between inside and outside at all parts of the room are shown in the first diagrams on Fig. 37.

Now consider the state of things when the small hole communicating between the inside and the outside air is not at floor level, but at a height h above the floor level, the temperatures being as before. This is shown in Fig. 38.

If there is at first any difference of pressure between inside and outside at the level of the hole, air must at once flow either in or out through the hole, until the difference of pressure on either side of the hole disappears. Not till then will equilibrium be established between inside and outside of the room. The pressure inside and outside will then be the same *at the level of the hole.*

In this case, assuming that the absolute pressure at the level of the hole is p_h, the pressure at any distance x below this level will be on the outside

$$p_h + x\ D_o$$

On the inside the pressure will be

$$p_h + x\ D_i$$

that is to say, below the hole the pressure outside will be greater than the pressure inside by the quantity

$$x\ (D_o - D_i)$$

Similarly, at a distance y feet above the level of the hole, the pressure inside will be greater than the pressure outside by the quantity

$$y\ (D_o - D_i)$$

Now consider the state of pressures when two holes are drilled through the wall, one at the ceiling level and the other at the floor level, the temperature being somehow or other maintained constant in the room, as shown in Fig. 40.

It is shown above that when only the bottom hole exists, and before the ceiling hole is made, the pressure at the ceiling level will be greater inside than outside. When the second hole is drilled, therefore, air will at once begin to flow out of the top hole, because of the difference in pressure. This will obviously cause air to flow in through the lower hole, because the abstraction of air from the room owing to the flow through the top hole, will at once reduce the inside pressure at the level of the lower hole below the pressure of the outer air. Air must therefore at once begin to flow into the lower hole. After a few seconds the amount of air flowing into the lower hole must be exactly equal to the amount flowing out of the upper hole, otherwise air would continuously either accumulate in or be abstracted from the room. Neither of these conditions is possible.

When both holes are open, therefore, a continuous flow of air will pass. At the level of the lower hole the pressure inside must be less than outside, otherwise air would not flow in, and at the upper hole the pressure outside is less than inside, otherwise air would not flow out. Therefore, there must be *some* level between these two points, at which the pressure inside is exactly the same as the pressure outside. This level is called the "*neutral zone.*"

If the two holes are of equal area, and if the temperature in the room is uniform, the level of the neutral zone will not differ appreciably from the mid-level between the two holes, that is to say, the level of the neutral zone will be $H/2$ above the floor, and $H/2$ below the ceiling. (The curve of absolute pressure is in reality a logarithmic curve, though we have assumed it a straight line. The difference between the two is very small for a small length.)

In this case, from the formula above, it will be seen that the difference in pressure at the ceiling level will be

$$H/2\,(D_o - D_i)$$

above the pressure outside. And the pressure at floor level must in the same way be

$$H/2\,(D_o - D_i)$$

below the pressure outside. All pressures being given in pounds weight per square foot.

Hence, referring back to formula, it will be seen that as the difference in pressure expressed in head of air column of the outside density is

$$\frac{\frac{H}{2}\,(D_o - D_i)}{D_o}$$

the velocity of the air out of the upper hole will be given by

$$v = \sqrt{2g \frac{\frac{H}{2}(D_o - D_i)}{D_o}} = \sqrt{g\,H\,\frac{(D_o - D_i)}{D_o}} \qquad \text{(see page 68)}$$

and the same will be the velocity into the lower hole, neglecting the difference in temperatures of the air passing through the two holes.

If the two holes have different areas, A_1 and A_2, the same *weight* of air will still flow in the same time through the larger as through the smaller, and the pressures in the room will adjust themselves so that

$$v_1\,A_1 = v_2\,A_2$$

or more strictly taking count of the difference of temperature

$$\frac{v_1\,A_1}{1 + a\,t_o} = \frac{v_2\,A_2}{1 + a\,t_i}$$

whence

$$\frac{v_1}{v_2} = \frac{A_2\,(1 + a\,t_o)}{A_1\,(1 + a\,t_i)}$$

But since

$$v_1 = \sqrt{2g\,\frac{p_1}{D_o}}$$

and

$$v_2 = \sqrt{2g\,\frac{p_2}{D_i}}$$

therefore

$$\frac{v_1}{v_2} = \sqrt{\frac{p_1\,D_i}{p_2\,D_o}} = \sqrt{\frac{p_1\,(1 + a\,t_o)}{p_2\,(1 + a\,t_i)}}$$

where p_1, p_2, are the respective *differences* of pressure within and without the room at floor and ceiling respectively, expressed in pounds per square foot.

Then, as

$$p_1 = x\,(D_o - D_i)$$

and

$$p_2 = y\,(D_o - D_i)$$

where x and y are the heights of the holes below and above the neutral zone respectively. Also

$$\frac{A_2\,(1 + a\,t_o)}{A_1\,(1 + a\,t_i)} = \frac{v_1}{v_2} = \sqrt{\frac{p_1\,(1 + a\,t_o)}{p_2\,(1 + a\,t_i)}}$$

or

$$\frac{A_2^2\ (1 + a\ t_o)}{A_1^2\ (1 + a\ t_i)} = \frac{p_1}{p_2} = \frac{x}{y}$$

In other words, the level of the neutral zone will divide the total height between the holes inversely as the square of the areas of the holes corrected for the difference in temperature.

If the holes are treated as plain orifices, the quantity of air passing out of them, assuming they are each of A square feet, will be

$$0.61 \times A \times \sqrt{gH\frac{D_o - D_i}{D_i}}$$

Apply this formula to a concrete instance.

Assume that the room has dimensions 25′ × 24′ × 14′, which are the dimensions of an ordinary class-room, designed for the accommodation of, say, 60 scholars, the cubic contents of the room being 8,400 cubic feet. Assume that the area of each hole is 1 square foot. The internal temperature of the room is 60° throughout, and the outside temperature is 32°.

Referring to Table V. it will be seen that the value of D_o is in these circumstances .0807, and of D_i .0764 pounds per cubic foot.

The density of the air will vary slightly according to the degree of humidity in the room. Assume that the humidity of the air is 60%. In this case the value of D_o will be about .0806. And D_i will be .0764.

The difference of pressure at ceiling level will be 7 × .0043 = .0294 lbs. per square foot, or 0.385 ft. of air column.

The velocity of the air will therefore be, assuming the same temperature at the two holes

$$\sqrt{32.2 \times 14}\ \sqrt{\frac{.0043}{.0764}} = 21.25 \times .233 = 4.95 \text{ feet per second.}$$

The total quantity of air passing through the openings will be

$$0.61 \times 1 \times 4.95 \times 3{,}600 = 10{,}800 \text{ cubic feet per hour}$$

and the total interchange produced by these two openings alone, due solely to the difference in temperature between inside and outside

$$\frac{10.800}{8400} = 1.30 \text{ interchanges per hour.}$$

It will be instructive and interesting to analyse in the light of these results some of the commonly applied rough rules, which are used by such bodies as Educational Authorities.

There is, for instance, a rule that in a room occupied by school children, the size of the ventilating apertures must be 4 square inches per child. According to the formula given above this would correspond in the given temperature conditions to an interchange of air per child of about

$$\frac{4}{144} \times 4.95 \times 3600 = 495$$

cubic feet per hour, when the temperature difference of 32° to 60° is maintained.

This is, of course, absolutely insufficient, as will be seen by reference to the tables on the Hygiene of Ventilation. The interchange would be even less when the external temperature was higher.

The above computation illustrates the imaginary condition of the ventilation under ideally exact conditions. These conditions being—

(1) Absolute stillness and uniformity of temperature of the air, both inside and outside, throughout the spaces concerned.
(2) Walls absolutely impervious to the passage of air.
(3) No other inlet or outlet than the two holes concerned.

Passage of Air through the Substance of Walls.

The actual conditions are, of course, always widely different from these. Brickwork or any kind of masonry is not impervious to air, but is quite porous, and any difference of pressure between two sides of an ordinary wall will always produce a flow of air through it.

In order that the subject may be understood, it is desirable to know how much air will pass through the material of which a wall of ordinary construction is made under a given difference of pressure, and this quantity is given in the following table.

The formula, which gives the amount of flow of air through *materials* of which walls are commonly made (the wall as built with mortar joints is much more porous that the brick itself), is as follows :—

$$V = \frac{A\,p\,\theta}{e}$$

where V is the volume of air passing per hour in cubic feet.
A is the area of the material in square feet.
p is the difference of pressure on the two sides of it in pounds per square foot.
θ is the porosity, or, in other words, the number of cubic feet of air which would be forced per hour through 1 ft. thick of material by a difference of pressure of 1 lb. per sq. foot.
e is the thickness of the wall in feet.

The values given by Lang, corrected for the change in units, for the *materials* of which walls are composed, are

Stone	.0065
Brickwork	.0105
Portland cement ...	.0072
Cast gypsum	.00215
Oak wood	.000367
Pine wood	0.053
Ordinary flooring ...	32 — 80 cubic feet per square foot per hour for a difference in pressure of 1 lb. per sq. ft.
Ordinary brick wall...	48 cub. ft. do. do.

whose porosity is .0105 under a pressure of one pound per square foot.

Now, since walls are porous, let us estimate what would be the quantity of air passing through an imaginary wall of a warm room, in which there are no regular openings for the purpose of ventilation, if the walls were composed of *solid* stone or brick without mortar or cracks, and all the doors and window crevices, etc., were perfectly tight.

Assume that the porosity of the floor material, which is perfectly tight, is represented by .053, and the walls, also perfectly tight, of solid material of porosity .0105, and the ceiling of material of porosity .007. Consider also that the roof is of the same dimensions as that quoted above, and assume, further, that the neutral zone is at the mid height of the room, that is to say, 7 feet above the floor.

The difference in air pressure at the ceiling will then be

$$7\,(0.0807 - 0.0764) = 7 \times .0043 = .0301 \text{ pounds per sq. ft.}$$

The same difference of pressure will exist at the floor.

The quantity of air passing through these two will therefore be as follows :—

$$\text{Floor (inwards)} \quad \frac{25 \times 24 \times 0.0301 \times 0.053}{{}^{1}/_{12}} = 11.5 \text{ cubic feet.}$$

$$\text{Ceiling (outwards)} \quad \frac{25 \times 24 \times 0.007 \times 0.0301}{{}^{1}/_{12}} = 15.2 \text{ cubic feet.}$$

The quantity of air passing into the room through the walls below the neutral zone will be found by the following :—

Consider a horizontal strip of wall $d\,x$ feet wide at a distance x feet below the neutral zone, of a room l feet long and w feet wide. The difference of pressure at this level is

$$x\,(D_o - D_i) \text{ pounds per square foot.}$$

Therefore, the amount of air passing in through this horizontal strip of wall will be, according to the formula

$$\frac{2\,(l + w)\,x\,(D_o - D_i) \times \theta \times d\,x}{e} \text{ cubic feet per hour.}$$

Integrating this between the level of the neutral zone (o) and the floor level (—$H/2$) we find that the formula giving the total quantity of air passing into the room below the neutral zone per hour is as follows :—

$$\frac{(l + w)\,(D_o - D_i)\,\theta\,H^2}{4\,e} + \frac{l\,w \times \theta \times \frac{H}{2}\,(D_o - D_i)}{e_1}$$

A similar formula will give the equal quantity of air passing outwards through the upper half of the wall and the ceiling.

The neutral zone will not be exactly at a height $H/2$ unless the porosity of the upper half is equal to that of the lower half. It will be seen, therefore, that the total interchange of air through the materials alone, apart from crevices, will be so extremely small as to be negligeable in practical calculations.

Now these conditions are also ideal conditions, which will never be found in practice. In practice there are numerous other crevices and apertures, not of microscopic dimensions, existing all over the room. Wherever there is any constructional want of continuity between the materials of which the bounding surfaces of the wall are composed, the magnitude of these apertures changes from day to day, owing to shrinkage and expansion of woodwork and other similar causes, caused by changes of temperature, dampness, and other conditions.

A further cause which will entirely upset any such calculations in practice is the external velocity of the wind, which will be considered later in the present chapter.

When wind impinges on any flat surface the relation between its velocity and the pressure exerted by it in pounds per square foot on a surface normal to the direction of the wind, is given approximately by the following formula :—

$$p = l\,\frac{v^2\,D}{2\,g} \text{ lbs. per square foot}$$

where $l = 0.75$ and $v =$ velocity in feet per second.

It will thus be seen that a velocity of wind equal to about four miles per hour will produce a pressure greater than the maximum difference of pressure which can exist, owing to the difference of temperature between the inside and the outside of a room of normal height.

Therefore, when any wind impinges irregularly and unequally on the two openings into a room it is obvious that the whole of these calculations are vitiated, and that the amount of air passing through the room is one which it is impossible to calculate. It is necessary to rely entirely on practical experience to obtain average figures. The figures quoted by different authorities differ so widely that the practical man is at a loss to know what to estimate.

On the one hand, some authorities assert that as much as two or three interchanges per hour should in practice be calculated on. Other authorities give the amount as from ½ to ¾ of an interchange. It is easy to see that between such limits as these no sort of agreement is possible. Most of these estimates are in fact mere irresponsible guesses. The difference of opinion is not difficult to understand, because the amount of air which does in fact pass through a room depends, as has already been shown, on variable factors.

A general rule commonly adopted and derived from practical tests, gives an interchange of from 1/200 to 1/40 of one interchange for every degree F. difference of temperature between inner and outer air, due to natural interchange alone in still air.

Further causes which give rise to wide differences are that the actual porosity of the material of the wall varies greatly, according to the amount of damp on it. A shower of rain will fill up the pores of the wall, so that the air interchange is greatly reduced. Certain kinds of wall paper fixed on the wall will practically destroy its porosity altogether, particularly papers of a varnished description, or glazed surfaces. If a wall is composed of glazed tiles, closely cemented together, the air interchange will also be practically nil.

A scientific investigation of the matter, therefore, shows that not only the amount of ventilation, but the amount of heat which must be supplied to a room varies so greatly that it appears almost useless to make any calculations at all. The actual fact is not so serious as this, because of the circumstance that many of the irregularities, to which attention has been called, neutralise one another to some extent.

For instance, when the amount of air passing into a room is largely increased, the radiators may be arranged to increase in efficiency at the same time.

The management of the boiler should be such that any changes of the weather, whether these are measured or not, can be compensated for by increased stoking. A good heating apparatus is a very flexible appliance, and in spite of all these variations, the broad effect is that when heating apparatus are calculated according to the standard formulæ based on these scientific considerations, and approved by practical experience, it is possible to attain in practice a balance between the heat required and the heat obtained. Although it is impossible to predict with certainty what will be the internal temperature in any other conditions than the ideal standard ones, which latter hardly ever exist in actual practice, yet if the allowance of heat is liberal enough, *i.e.*, if the co-efficients are great enough, it is found that the heat calculated from them is sufficient to satisfy a reasonable interpretation of the imaginary conditions for which the calculations are made.

If the ventilation in a room is found inadequate, the windows can be opened wider, and at the same time increased stoking on the boiler gives the additional supply of heat required.

The question of the neutral zone, and the questions allied to it have an important bearing on ventilation in respect of draughts, and particularly does this apply to forced systems of ventilation.

The above facts show that the amount of air which does in fact pass through the walls of a building when there are no special means of ventilation adopted cannot be calculated with any approach to practical accuracy.

Haldane's Experiments on Air Interchange.

It is therefore very desirable to note the results of experiments conducted by a reliable authority on this point. Such experiments have been made by Haldane for the Report of the Departmental Committee on the Use of Water Gas. (See table in the Appendix.)

The method adopted in these experiments consisted in introducing a measured quantity of carbonic acid into the air of the room by means of the combustion of measured weights of candle of known composition. Continuous analyses were then made of the air of the room, and the quantity of incoming and outgoing air calculated, presumably on the method explained in the last chapter. The results of these experiments are of great interest to the heating engineer as illustrating the amount of ventilation which does take place when no special means of ventilation are present.

Experiments on eight different buildings are reported, the observations being taken on the same building in widely varying conditions. They therefore form a very useful guide to the heating engineer. A resumé of the results of these experiments is given in the table in the Appendix.

It will be seen that the amount of air passing into and out of all the rooms varied greatly, according to the conditions. That the chief influence in producing interchange was the strength of the wind outside. A large difference was also produced by differences in temperature between inside and outside.

It will be seen, too, that to abstract any regular and definite rules from the results of these most careful experiments is difficult, if not impossible, and that general conclusions only can be formed.

It will be seen, too, from the results of these experiments that any specific and exact calculations on such a point as the rate of interchange of air, by what is called natural ventilation, are quite superfluous.

Analysing these results generally it will be seen that the larger a room the lower will be the rate of interchange by penetration of air through the walls. The superficial area of the walls, roof, floor, etc., only increases as the square of one linear dimension, whereas the capacity increases as the cube of the same dimensions.

Thus, an increase of 2 to 1 in one linear dimension causes an increase of wall area, etc., of 4 to 1, and an increase in space of 8 to 1.

It will be seen, too, that the paramount influence for producing interchange is the velocity and direction of the wind, and that a gentle breeze of 10 miles an hour has in practice far more effect than a difference of temperature of 20°.

That the effect of burning a bright fire in the grate is very beneficial and increases the room ventilation as much as 10 times. The mere closing or opening of the flap of a grate without a fire makes a difference of 2 to 1 in the ventilation.

In the larger halls, even in cases where the roof is very leaky, natural interchange alone produces one renewal of air in three hours. The experiments indicate, too, that for small rooms, provided always there is an open chimney and an air space of no less than 1,000 cubic feet per person, the ventilation may suffice without opening the window, though this method is not to be recommended.

With a fire in the grate the ventilation may be sufficient for only 400 cubic feet of space per person. The larger the room, however, the greater becomes the need for special means of ventilation, and in large and crowded rooms, it is impossible to provide adequate ventilation otherwise than by the use of fans.

The same authority has analysed the air in many public elementary schools.

The average impurity of the air without fans was found to be 18.6 during the winter months, with an average of 186 cubic feet of air space per child, and 15.450 cubic feet per room (Carnellie, Haldane, and Anderson, "Phil Trans.," 1887).

It may be stated generally that no crowded room can have a quantity of air passed through it sufficient to supply to each person an amount recognised as sufficient by any medical authority without the provision of mechanical means of ventilation. The means employed is almost universally that of a fan, either of the centrifugal or propeller type, or less well-known types, though other means are of course sometimes employed also.

The object to be attained is to design the position of the neutral zone in a room such that no inconvenient draughts are caused. Variations in the position of the neutral zone are brought about, as has been already explained, by variations in the size and positions of the inlet and outlet in what is called the "natural system" of ventilation, by which is to be understood the mere provision of openings.

In cases where the inlet and outlet from a room are not mere openings through the wall, but in which these openings are connected to flues running above or below the room itself, great variations in the ventilation are, of course, produced according to the temperatures maintained in the flues and in the room.

The Position of the Neutral Zone.

The position of the neutral zone and its bearing on the question of draughts will be best understood by considering the following diagrams :—

Let Figs. 37 to 40 represent a vertical section through a room, in which the temperature is t_i, the outside temperature being t_o, and assume that openings are provided making connection between the interior and exterior of the room.

In the case where there is one opening at floor level only, as in Fig. 37, it has been shown that the pressure at the floor level inside the room is the same as outside, and that the difference of pressure increases inside the room roughly in proportion to the height.

The state of pressure, therefore, will be shown by a sloping line, such that the line $B\ C$ is equal to

$$H\ (D_o - D_i)$$

and the difference of pressure at any height will be shown by the distance between such line and the line representing the walls of the room. The slope of the line is determined by the difference of temperature. The greater the difference of temperature the more the line slopes away from the perpendicular.

Fig. 38 represents the state of relative pressure when the opening is provided in the middle of the wall, and Fig. 39 the case if there is one opening at the ceiling only. When the temperatures are the same the line $A\ C$ is parallel to $A_1\ C_1$ and to $A_2\ C_2$ or $A_3\ C_3$.

In this last case the difference of pressure at the ceiling and also at the floor is, as has been proved,

$$H/2\ (D_o - D_i) \text{ pounds per square foot.}$$

The corresponding diagrams, when the inside of the room is colder than the outer air, are shown in Figs. 41 to 44.

Now consider the case of two rooms close together, having a wall between them, and having the same temperature.

Assume first that the level of the neutral zone is the same in each case

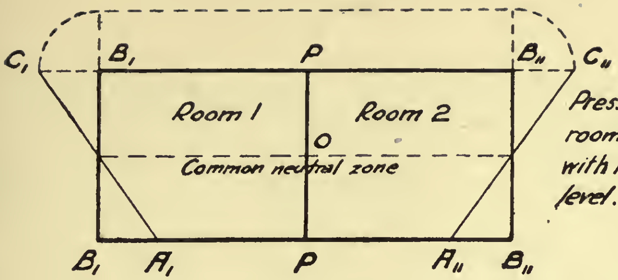


FIG. 45

Pressures in two adjacent rooms at same temperature with neutral zones at same level.

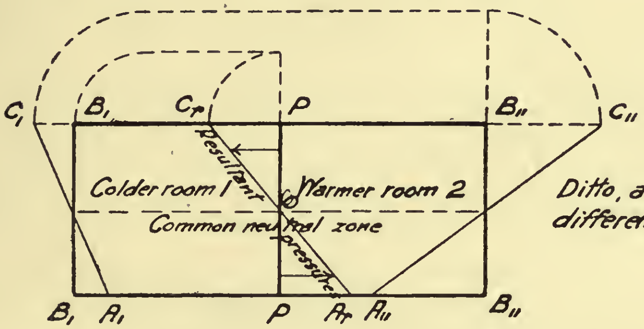


FIG. 46

Ditto, as above, but at different temperatures.

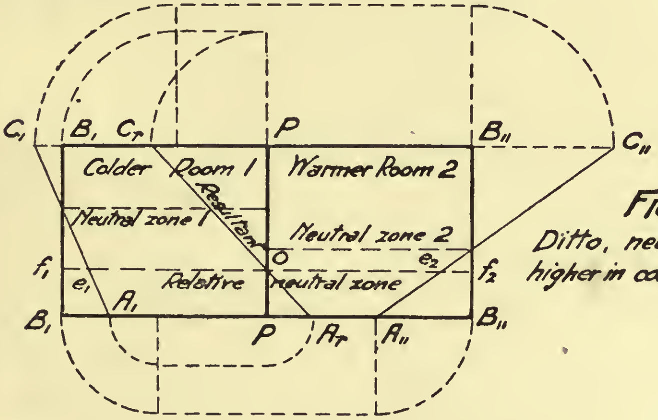


FIG. 47

Ditto, neutral zone higher in colder room

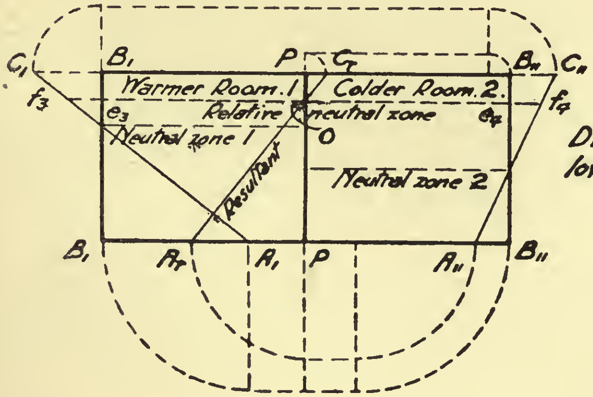


FIG. 48

Ditto, but neutral zone lower in colder room.

and the temperatures are the same. In this case $A' \ C'$ represents the difference of pressure in one room, and $A'' \ C''$ in the other.

Since the temperatures are assumed to be the same, the line $A' \ C'$ has the same inclination as $A'' \ C''$, and it is obvious, therefore, that the pressure at any given height is the same in the two rooms, and therefore any opening between them at any level will not be the cause of a draught.

If, however, though both the neutral zones are at the same level, the temperatures in the two rooms are different, then the two lines representing the pressure will have a different slope, as in Fig. 46.

In this case there will always be a greater pressure in the warmer room above the neutral zone than in the colder room, and below the neutral zone the pressure in the colder room will be greater than the pressure in the warmer. Clearly $B'' \ C'' - B' \ C' = P \ C_r$. The resultant pressures between the two rooms are represented by the line $A_r \ C_r$ and $A'' \ B'' - A' \ B' = P \ A_r$.

Below the neutral zone, therefore, there will be draughts from the colder room to the warmer through any apertures which exist, and above the neutral zone from the warmer to the colder. The movement of the air, in fact, will be such as to tend to equalise the temperature between the rooms.

Now consider Fig. 41, which shows the state of two adjacent rooms at which the level of the neutral zone is not the same.

Assume first that the level of the neutral zone is higher in the colder room. In this case the inclination to the vertical of the line, which represents the state of pressure in the warmer room, will be greater than that in the case of the colder room.

We may now draw a line showing the resultant pressure between the two rooms, which will also be a sloping straight line, and the resultant neutral zone between the two rooms will be at such a level below the actual neutral zones that at this level the pressures in the two rooms are the same, that is to say, that the line $e_1 \ f_1$ is equal to the line $e_2 \ f_2$.

The resultant slope will represent the *difference* between the slopes of the two other lines. The construction indicated in this diagram shows how the required slope may be determined geometrically.

It will be seen on examining this diagram that the lower the temperature in the cooler room the more nearly does the level of the resultant neutral zone coincide with the absolute neutral zone of the warmer room, until when the lower temperature is the same as the outside air the two neutral zones coincide.

As the temperature of the colder room increases, assuming its absolute neutral zone remains at the same level, the lower does the resultant neutral zone become. At a certain temperature it sinks below the level of the floor.

At the point where the temperatures in the two rooms are the same the line of relative pressures is parallel with the partition line between the two rooms, that is to say, the difference of pressure is the same at all levels. The resultant neutral zone may then be supposed to be at an infinite distance away.

When the temperature in what was the cooler room rises above that of the warmer room, we have the case where the level of the neutral zone is lower in what is now the colder room. In this case the resultant neutral zone between the two rooms will be above the level of the neutral zone in the warmer room, at such a point that the line $e_3 f_3$ is equal to the line $e_4 f_4$, for at this point the pressure in the two rooms is evidently the same, being in each case above the atmospheric pressure at the same level by the same amount. This state of things is indicated in Fig. 48.

In this case the line of resultant pressures will be $A_r C_r$, passing through the point 0 and the point A_r where $P A_r = A' B' - A'' B''$. As the left room becomes still warmer the point 0 sinks towards the neutral zone of the warmer room.

These matters are of importance when considering the relation between the ventilation of adjacent rooms used for widely different purposes. For instance, between a restaurant and a kitchen attached to it.

In a case like this it is very important that no air should pass from the kitchen to the restaurant, so that there shall be no smell of cooking in the latter. To avoid this the pressure at *all heights* should be greater in the restaurant than in the kitchen. The temperature in the kitchen will probably be considerably higher than in the restaurant. Unless artificial means are adopted for raising the level of the neutral zone in the kitchen, some of the air therefrom will pass into the restaurant through any opening above the level of the resultant neutral zone, which would be very objectionable.

In such a case the absolute neutral zone in the kitchen must be raised so high that the resultant neutral zone between the kitchen and the restaurant must be above the level of the ceiling of the kitchen. It is not necessary, as is sometimes asserted, that the absolute neutral zone in the kitchen should be above the ceiling, but merely that the *resultant* neutral zone should be above the ceiling of the kitchen, which is a much less stringent requirement, as may be seen above. The slope of the line may be calculated from the formula

$$p_i - p_o = H (D_o - D_i)$$

Pressure in Vertical Air Pipes.

In order that the conditions for the ventilation of rooms by heated upcast flues may be properly understood, it is necessary to consider the variation of pressure in a vertical flue through which a current of air is passing. The general principle on which such a flue works has been explained in Chapter VII.

It has also been shown above that a ventilated room having an air inlet at the floor, and outlet at the ceiling, is really an upcast flue of large sectional area, and having small inlets and outlets. In this case the *mean* upward velocity of the air within the room is so small as to be negligeable, though there may, of course, be local upward currents of considerable velocity.

Imagine that the horizontal dimensions of such a room gradually contract until the room becomes a mere flue. It is evident that when the horizontal dimensions of the room contract so far that the cross sectional area of the room is the same as the area of the inlet and outlet, the room becomes a plain upcast flue. The variations in internal and external pressures, explained on the previous pages, will apply to such a flue only when the top and bottom openings are small compared to the sectional area.

When the sizes of the openings are comparable with the sectional area, the frictional resistance within the flue, due to the relatively high velocity of the air, disturbs these conditions.

It will be necessary, therefore, to compare the pressures within a warm flue, up which a current of air is passing, with the pressures of the atmosphere at the same level.

Let Fig. 49 represent such a flue, and assume that it is closed at the top and bottom by adjustable dampers, either of which can close the top or bottom of the flue entirely, or open them to any desired extent up to full area of the flue.

Assume first that the bottom damper is wide open, as shown in Fig. 49. When the top is closed we shall have the same condition of things as was explained at Fig. 37. The pressure at the bottom of the flue will be the same as at the same level in the outside air, and the relative internal pressure will increase gradually towards the top, as is shown by the diagram.

Now let the upper damper be slightly opened, the temperature in the flue being always maintained constant by means of a battery of no resistance fixed at the bottom.

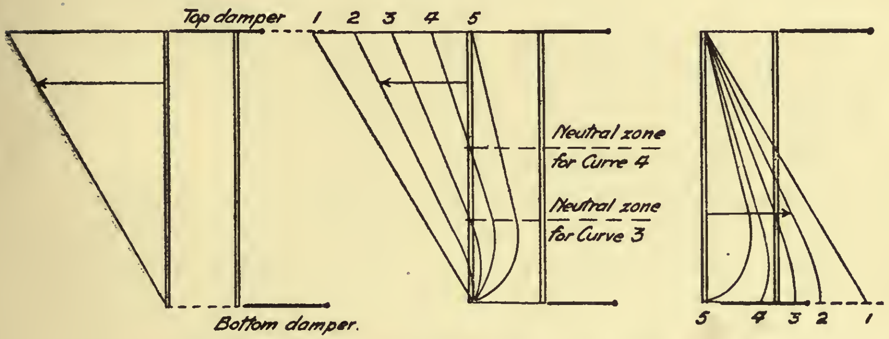
The opening of the upper damper will obviously cause an outflow of air, because the pressure on the inside is greater than on the outside. When the opening is very small its presence will only affect to a very slight extent the pressure in the upper part of the inside of the tube. It will cause, as a whole, a slow movement of air up the tube, the reduction in pressure at the top, such as it is, will be due to the head necessary to set the air in motion at this small velocity, and to overcome the slight frictional resistance.

The wider the damper is opened the greater will be the reduction of pressure at the top, until, when it is wide open, the pressure at the top will be substantially identical with that of the atmosphere. These successive changes in pressure, as the upper damper is opened wider and wider, are shown by Fig. 50, where curve 1 represents the pressures all the way up the tube, when no air is flowing; and curves 2, 3, 4, 5, represent the pressure as the damper is opened wider and wider.

Whatever the opening, it is important to observe that the weight of air passing any section of the flue per second must be constant, otherwise air would either accumulate in or be abstracted from the flue continuously, which is impossible.

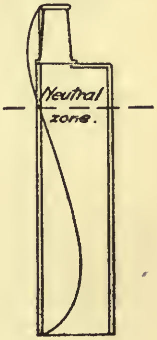
It is important also to observe that the higher the air rises the lower is its absolute pressure, and therefore the greater is its volume, but to a comparatively slight extent. The velocity, therefore, at the upper part of the flue must be greater than at the lower part, because of the continual expansion of the air owing to the relief of pressure on it.

FIG. 49. FIG. 50. FIG. 51.



DIAGRAMS ILLUSTRATING PRESSURES IN A WARM FLUE.

FIG. 52.



EFFECT OF CHIMNEY POT.

The friction therefore becomes greater in the upper part of the flue than in the lower.

One other consideration must also be borne in mind. It has been already pointed out that if air or any other fluid flows in a continuous stream between any two sections of a uniform flue, the pressure across the first section must be greater than the pressure across the second section, by just so must as is necessary to overcome the friction of the flue between these two points. Just outside the flue at the bottom the air is at rest, just inside the flue at the bottom it is moving with some considerable velocity. It is evident, therefore, that at the bottom the pressure inside the flue must be less than the pressure outside. Actually at the opening to atmosphere the pressure must be the same as the atmospheric.

It is therefore evident that the form of the curve of relative pressures must be as shown.

It is instructive also to consider this matter by assuming that the operation is started with the upper damper wide open and the lower one closed. In this case the curve of relative pressures between the inside and outside of the tube is shown on Fig. 51 (1). As the lower damper is gradually opened, the pressure inside gradually increases in the manner shown by the successive curves, Nos. 1, 2, 3, 4, 5, in Fig. 51, until, when the damper is wide open, the curve must have the same form as the last curve, 5, in Fig. 50.

A consideration of these curves will enable the operation of a vertical upcast shaft to be understood. It is to be observed that the generation of the velocity in the rising column of air takes place at or near the bottom of the flue. Once the momentum of the air has been generated, no more pressure, apart from the inevitable friction of the air in the flue, is necessary merely to maintain that momentum. But even though the momentum is attained there must be a continual loss of *circulating* pressure all the way up the flue in order to maintain the velocity.

It is for this reason that the curve of relative pressures in an upcast flue corresponds more nearly to the line of pressures with the bottom damper closed than to that with the top damper closed. The greater part of the apparent resistance of a short flue is due to the inertia of the air. In a long flue a smaller proportion of the total motive power generated is absorbed in setting the air in motion than in a short flue. If, therefore, the line of relative pressures with the air in motion (5) is compared with the line of pressures when the bottom damper is closed (curve 1 in Fig. 51), it will be inclined to that line at such an angle as will represent the rate of absorption of pressure due to friction. At the bottom of the column it will curve steeply outwards in such a way that it becomes zero at the lower opening to the atmosphere.

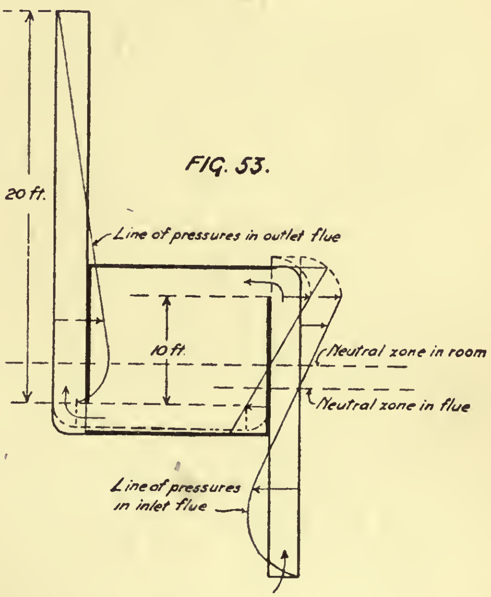
When, however, any kind of resistance, such as a chimney pot or other obstruction, is introduced in the upper part of the flue, so that the air has not a free course at undiminished velocity within the flue, the necessary increase in velocity due to the restricted area can only be secured by an enhanced pressure inside the flue at the top. In such a case there must always be a neutral zone within the flue itself, that is, a point where the line of relative pressures crosses the base line, and the greater the resistance at the top the lower will this neutral zone lie. These neutral zones are shown dotted in Fig. 52.

Velocity in a Vertical Flue.

The method of calculating the velocity which will be set up this vertical flue is to calculate what is the total motive power or circulating pressure. This depends solely on the height. The velocity V of the air will then be given by the solution of the equation :—

$$\frac{H\,(D_o - D_i)}{D_i} = \frac{V^2}{2\,g}\left(1 + \rho\,\frac{l\,c}{a} + \Sigma\,\zeta\right)$$

The value of ζ being taken from the given table according to the kind of walls which the flue has.



20 ft.
Line of pressures in outlet flue
10 ft.
Neutral zone in room
Neutral zone in flue
Line of pressures in inlet flue

FIG. 53.

In order that this formula may be readily employed, the appended diagram (Fig. 54) has been plotted, showing the velocities theoretically attainable in an upcast flue of different heights at different temperatures.

To calculate the amount of air which will pass through a given system of flues at known temperatures the same formula must be employed. For instance, calculate the quantity of air which will pass through a room provided with ventilating flues, such as are shown in Fig. 53. Here the total motive power is the difference in weight of a column 25 feet high at 40°, and the weight of a composite column consisting of 15 feet at 80°, 20 feet at 75°, less 10 feet at 60° is as follows :—

$$D_{40} = 0.0863 \times \frac{461}{501} = 0.0795$$

$$D_{60} = 0.0863 \times \frac{461}{521} = 0.0765$$

$$D_{75} = 0.0863 \times \frac{461}{436} = 0.0742$$

$$D_{80} = 0.0863 \times \frac{461}{541} = 0.0736$$

Circulating pressure =

$$25 \times 0.0795 + 10 \times 0.0765 - 20 \times 0.0742 - 15 \times 0.0736$$

$$= 0.166 \text{ lbs. per sq. foot}$$

$$= \frac{0.166}{0.0742} = 2.24 \text{ feet of air column at } 75^\circ$$

This calculation shows that the total circulating pressure is 2.24 feet of air column at 75°. If this sets up a velocity V in the first flue the head lost in that flue will be represented by

$$\frac{V^2}{2\,g}\left(1 + \rho\,\frac{l_1\,c_1}{a_1} + \Sigma\,\zeta_1\right) \text{ feet of air column at } 80^\circ$$

The corresponding velocity in the outlet flue is given by

$$V\,\frac{a_1}{a_2} \times \frac{461 + 75}{461 + 80} = 0991\,\frac{V\,a_1}{a_2}$$

which will set up a total friction of magnitude

$$\left(\frac{0.991\,V\,a_1}{a_2}\right)^2 \times \frac{1}{2\,g}\left(1 + \rho\,\frac{l_2\,c_2}{a_2} + \Sigma\,\zeta_2\right)$$

in feet of air column at 75°

There will be practically no friction due to the air passing through the room, because the area is so large that the velocity will be practically zero.

The total friction in terms of V will be found by adding the values of the two expressions together and equating to the total circulating head.

The quantity of air will be found by solving the equation for V, and multiplying by the area of the larger flue.

This result will, of course, be altered by any wind impinging on the building, which on the one hand may accelerate, and on the other may retard the interchange, according to the direction of the wind.

The upward motion of the air, of course, implies the existence of a constant supply of power. This mechanical power is actually derived from the automatic conversion from the heat employed in warming the air into mechanical energy by the expansion of the air.

The total amount of mechanical energy brought into existence by heating a given volume of air has already been discussed in Chapter V., in which it is shown that of the total amount of heat used in heating air at a constant pressure, only a portion is actually employed in heating the air, the remainder being employed in pushing back the pressure of the atmosphere. It is this energy which is the source of the motive power which overcomes the friction.

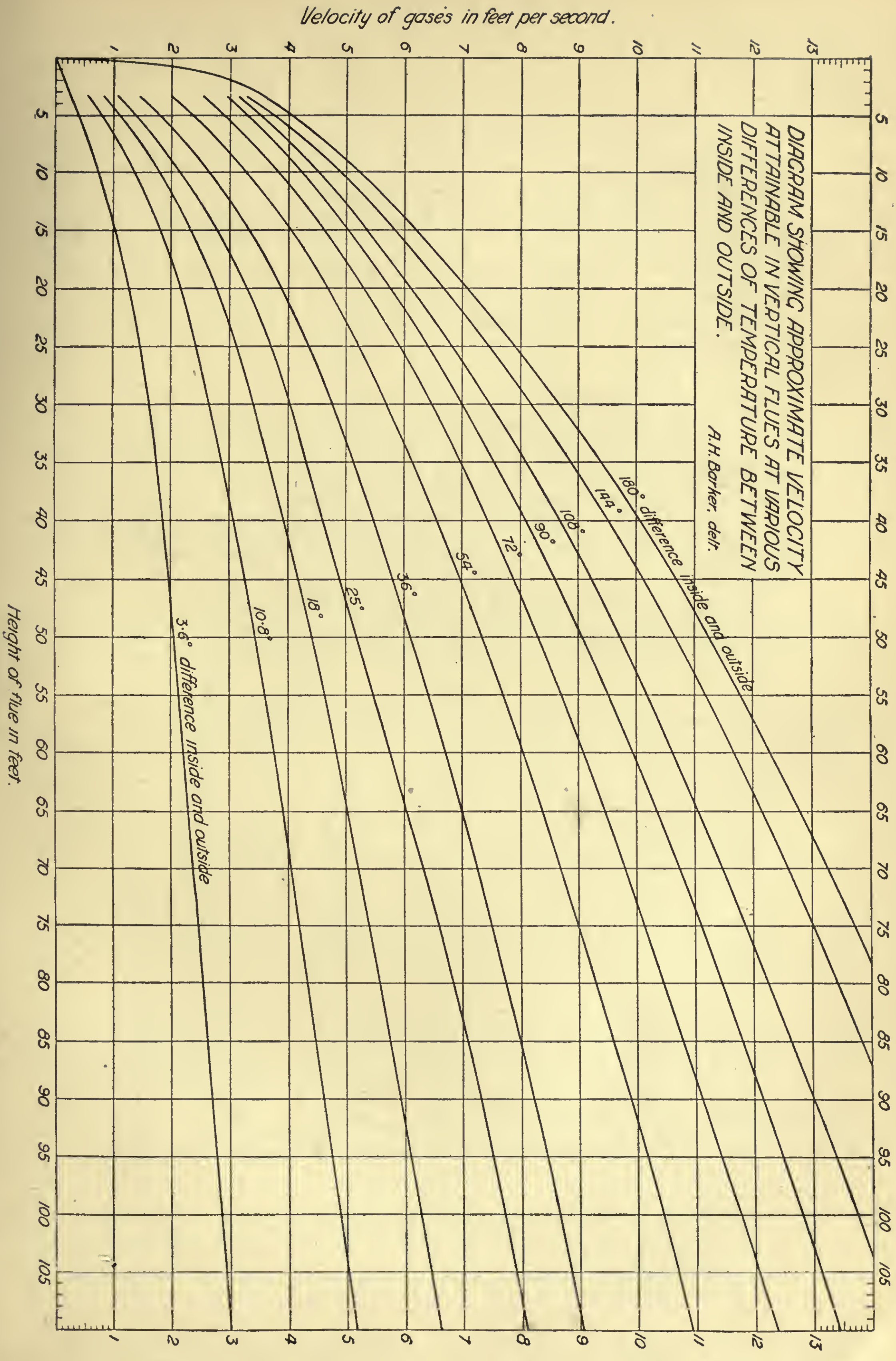

Velocity of gases in feet per second.
DIAGRAM SHOWING APPROXIMATE VELOCITY ATTAINABLE IN VERTICAL FLUES AT VARIOUS DIFFERENCES OF TEMPERATURE BETWEEN INSIDE AND OUTSIDE.
A.H. Barker. delt.
180° difference inside and outside
144°
108°
90°
72°
54°
36°
25°
18°
10·8°
3·6° difference inside and outside
1 2 3 4 5 6 7 8 9 10 11 12 13
5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100 105
Height of flue in feet.

FIG. 54.

The calculation of the position of the neutral zone in a room provided with flues of this kind may be made as follows :—

The velocity of air up the flues is known from the previous calculation.

The pressure within the extract flue at the top is the same as that of the atmosphere at the same level.

The relative pressure inside the flue at the bottom may then be calculated as

$$H\ (D_i - D_o) - h_f$$

where h_f is the pressure necessary to overcome the friction in the flue.

The pressure in the room at this level must therefore be such as to get up the calculated velocity V forcing the air into the extract flue. The known temperature of the room now gives the slope of the line of relative pressures and therefore the level of the neutral zone.

A similar calculation may be made, starting from the inlet to the fresh air flue, which may be reckoned as at the same pressure as the atmosphere.

These calculations are not of sufficient practical importance to justify their treatment at greater length in the present volume. It is important, however, that the principle involved should be thoroughly understood.

CHAPTER XI.

FANS.

BEFORE commencing the discussion of fans it will be well to summarise the formulæ relating to the power theoretically necessary to propel air against definite heads or with a definite velocity.

Q cubic feet of air weighing D pounds per cubic foot cannot be forced against a pressure of H feet of air column without a minimum expenditure of work equivalent to QDH pounds. The work necessary is clearly that required to lift QD pounds of air through a height H feet, or, in other words, to force Q cubic feet against a pressure of HD pounds per square foot.

If the pressure is expressed as h inches of standard water column it must be observed that if D is the density of air in pounds per cubic foot and 62.4 is the density of water in the same units, then water is $\frac{62.4}{D}$ times heavier than air, and h'' water $= \frac{h \times 62.4}{12\,D}$ feet of air column.

Now for ordinary purposes D may be taken to be 0.0761 lbs. per cubic foot, which is the actual density of dry air at 60 and 14.7 lbs. per square inch. Hence water is $\frac{62.4}{0.0761} = 820$ times heavier than dry air at 60°.

Hence, 1″ of water column = 820 inches or 68.3 ft. of air column, or 1 foot of air column $= \frac{12}{820} = .0146$ inches of water column.

Hence, to force Q cubic feet of air against h'' of water column necessarily involves an expenditure of energy

$$Qh \times 68.3 \times 0.0761 = Qh \times 5.2 \text{ foot pounds.}$$

If this quantity of air, Q cubic feet, has a velocity v communicated to it, the quantity of energy necessarily absorbed is

$$\frac{Q \times 0.0761 \times v^2}{64.4}$$

foot pounds $= Q\,v^2 \times 0.00118$ foot pounds.

Hence, if Q cubic feet of dry air at 60° F. and 14.7 lbs. per square inch are delivered at a velocity v against a head of h'' water gauge, the *minimum* quantity of energy necessary is

$$Q\,(5.2\,h + .00118\,v^2) \text{ foot-pounds.}$$

If these figures are to be corrected for alterations of temperature, barometric pressure and moisture, the following formulæ are useful :—

The weight of one cubic foot of air at other temperatures and pressures, taking the moisture into account, is as follows :—

$$\frac{1.33}{T}\,(B - 0.375\,b)$$

where B is the height of the barometer in inches of mercury and b is the pressure of the vapour in the atmosphere in inches of mercury, T being the absolute temperature.

If it is desired to take these variations into account this value of the density must be substituted for the above given figure 0.0761 in all the above calculations.

Taking account of the height of the barometer only, if Q cubic feet of air at absolute temperature T and with a barometer at B'' are discharged per minute at a velocity V ft. per second, the horse-power in the air discharged will be

$$\frac{V^2\,Q\,B}{T} \times 0.00003756$$

Similarly the horse-power absorbed when Q cubic feet of air per minute are delivered against a water gauge of h'' of water is found as follows :—

$$1'' \text{ water} = \frac{62.4}{12 \times D} \quad \text{or} \quad \frac{62.4}{12 \times \dfrac{1.33}{T}\,B} = 3.91\,\frac{T}{B}$$

feet of air column.

Hence, horse-power =

$$\frac{Q \times 1.33\,\dfrac{B}{T} \times 3.91\,\dfrac{Th}{B}}{33{,}000} = Qh \times 0.000158$$

Variations in the temperature and barometric pressure do not affect this formula. The work done in forcing a definite volume against a definite pressure depends, of course, only on the volume and the pressure and is independent of the density of the fluid, because the product of the density and the height of air column equivalent to a given pressure is always constant.

Hence, combining the above two expressions, the minimum horsepower required when Q cubic feet per minute of air at absolute temperature T, and pressure B'' of mercury are discharged at a velocity V against a water gauge h'' of water is

$$Q\left(\frac{V^2 B}{T} \times 0.00003756 + h \times 0.000158\right)$$

General Explanation of Working of a Centrifugal Fan.

It has already been shown that the pressure in a channel through which air is flowing varies with the variations of velocity. If air is moving in a flue with velocity v_1 which is *gradually* reduced to v_2 by a gradual enlargement of the flue, some of the kinetic energy of the air is converted into what is mis-named "pressure energy," and the pressure head in the flue increases according to the law

$$\frac{v_1^2 - v_2^2}{2g} = H_2 - H_1 \text{ feet of air column.}$$

If then at a certain part of an air conduit it is possible to introduce into an expanding portion of the flue a continuous stream of air at a certain pressure and moving along the flue with a considerable velocity, so that the quickly moving air is slowly reduced in velocity, an additional pressure will be developed in front of the stream. This pressure so generated may be used for driving the newly introduced air along an extension of the conduit.

It is precisely this which a centrifugal fan accomplishes. The small part of the flue in which it is desired to raise and maintain a velocity of air, is wrapped round the circumference of a wheel provided with vanes of such a shape that when the wheel rotates, the air between the vanes is swirled radially outwards by centrifugal force, and is driven off from the outside circumference of the wheel in a direction inclined outwards at a slight angle to the tangent to the fan wheel. The rotation of the fan wheel produces and maintains a continuous stream of air, each ray of which maintains a direction symmetrically disposed with regard to the axis of the fan, the velocity of each ray being roughly constant all round the wheel.

The general construction will be understood from Fig. 55.

The "volute," as the trumpet-shaped portion of the flue wrapped round the fan wheel is called, picks up all these rays and forms them into one stream of air, which regularly increases in cross section in such a way that the velocity in the volute is approximately constant. Ultimately, after the volute has passed completely round the wheel, it delivers the massed stream of air into the fan outlet, which is called the "chimney," and which should taper outwards gradually and open into the flue into which the air is delivered, without any sudden change of section.

A centrifugal fan consists essentially of (1) the wheel, provided with radial vanes of various shapes, differing according to the purposes for which the fan is designed or according to the idea of the designer; (2) the diffuser, which is a cylindrical chamber into which the air from the tip of the fan is delivered, its purpose being, as its name implies, to allow the air to diffuse itself uniformly and the different velocities of the air from different parts of the fan tip to equalise themselves and to become slightly reduced before the air is delivered into (3) the volute, as the expanding spiral collecting channel around the entire fan is called.

For economy in construction, however, most fans are made without any diffuser, the air delivering direct from the fan blades into the volute, as in Fig. 56.

The volute and the diffuser are sometimes omitted where the air is delivered direct into the atmosphere, that is, when the fan is used for sucking air from a flue, and not for delivering air into a flue under pressure. When this is done the fan works entirely on centrifugal force.

An additional and a very useful adjunct to a fan is an expanding "chimney" in which the velocity of the air in the volute is gradually reduced, so as to increase the pressure against which the fan will work.

The Theory of a Perfect Centrifugal Fan.

There are thus two sources of the pressure which the working of a centrifugal fan produces in the flue. The first source is that the motion of the vanes delivers air into the chimney at a high velocity v_1, the pressure being generated by the gradual reduction of this velocity to v_2, owing to the enlargement of the flue. The amount of the rise in pressure has been already shown to be

$$H_2 - H_1 = \frac{v_1^2 - v_2^2}{2g} \text{ in feet of air column.}$$

If the chimney could be expanded so far that v_2^2 was practically negligeable compared to v_1^2 we should have

$$H_2 - H_1 = \frac{v_1^2}{2g}$$

in which case the whole of the kinetic energy of the air would be utilised.

The second cause of rise in pressure, due to the fan, is the centrifugal action on the particles of air, which is caused by the rotation of the vortex of air within the fan wheel.

When a stone is swirled round at the end of a string, its centrifugal force exerts a pull on the string. In just the same way, when a particle of air is caused to swirl round an axis, an outward force acts upon it, which tends to force it away from the axis.

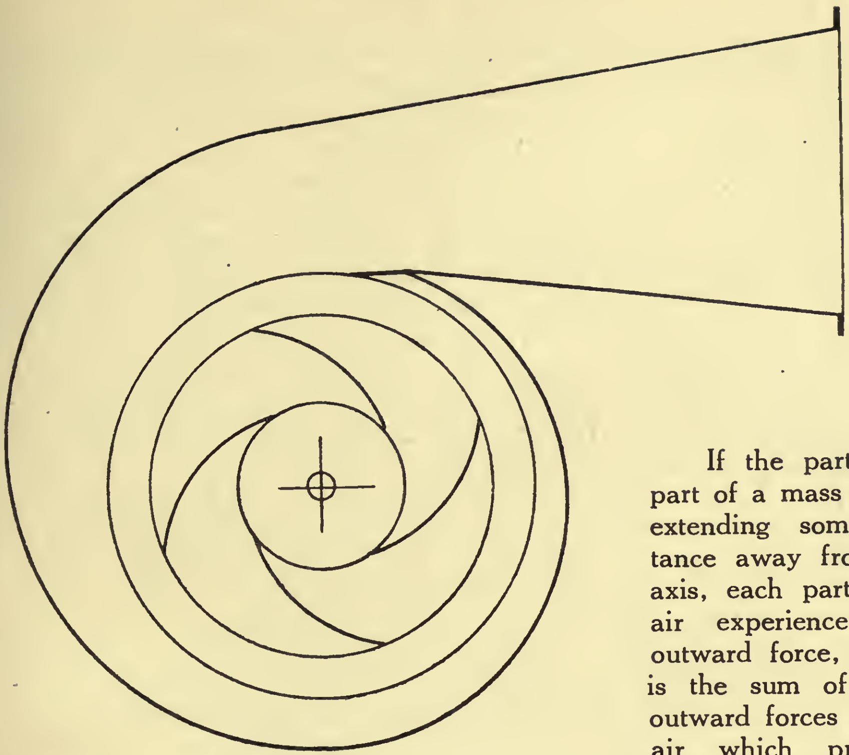


FIG. 55.

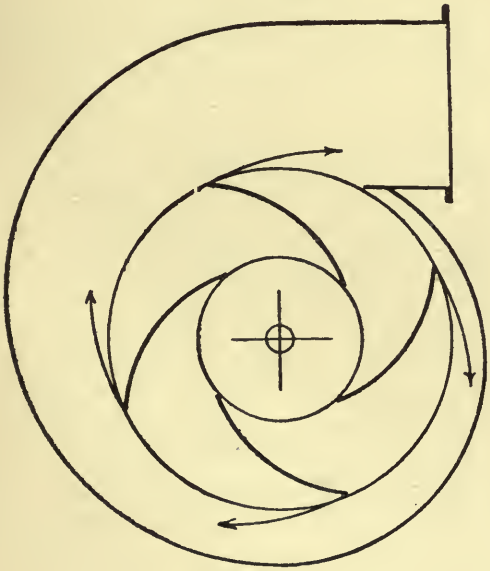


FIG. 56.

If the particle is part of a mass of air, extending some distance away from the axis, each particle of air experiences this outward force, and it is the sum of these outward forces on the air which produces part of the pressure due to the working of a centrifugal fan.

We shall now calculate what the magnitude of this pressure is, on the assumption that the fan is perfect.

For the purpose of this calculation the shape of the vanes makes no difference. The one assumption is that all the air within the fan wheel is carried round with the same speed as the vanes. For the purpose of eliminating any complication that may arise owing to the radially outward movement of the air between the vanes, we shall assume the outlet closed,

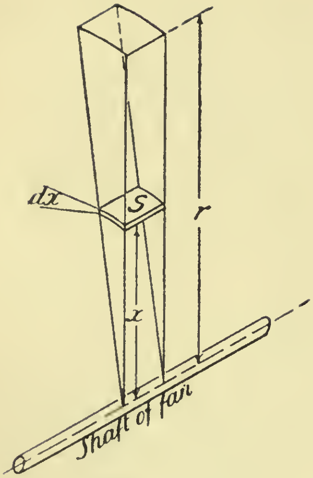


FIG. 57.

so that the fan is creating a stationary vortex of air moving round at a constant angular velocity, and that the velocity of this air has no radial component.

Consider any small part of an imaginary cylindrical layer of this air within the vortex, the axis of the cylinder being the axis of rotation, and the radius of the cylinder being x feet. The thickness of the cylindrical layer is dx, and the small part of the area under consideration, measured along the circumference of the cylinder is s square feet. The volume of this small mass of air is therefore (see Fig. 57)

$$s\ dx \text{ cubic feet,}$$

and where D is the density of the air in pounds per cubic foot, its weight is

$$s\ dx\ D \text{ pounds.}$$

Assume that the angular velocity of the wheel is ω radians per second, then the outward force acting on this small mass of air will be

$$\omega^2 \times x \times s \times dx \times D \text{ poundals (see page 9).}$$

Obviously, therefore, since by assumption this outward force does not generate any motion in the air, this expression represents the difference in the *force* on the outside and on the inside of the layer. Where df is this small difference we have

$$df = \frac{\omega^2 \times x \times s \times dx \times D}{g} \text{ in pounds weight.}$$

Dividing both sides of this equation by s we have the difference in *pressure* reckoned in pounds weight per square foot on this small area.

Since

$$\frac{df}{s} = dP$$

$$dP = \frac{\omega^2\ x\ D\ dx}{g} \text{ pounds weight per sq. ft.}$$

or dividing by D, since $\frac{dP}{D} = dH$ the difference in head of air

$$dH = \frac{\omega^2\ x}{g}\ dx$$

Integrating this value between the limits R and r we obtain the whole value of the increase in pressure between these radii

$$H = \frac{\omega^2 r^2 - \omega^2 R^2}{2g}$$

If, therefore, the whole of the air from the centre of the axis to the outer tip is rotating along with the wheel, that is, when $R = o$, the difference in pressure due to the rotation of the fan is

$$\frac{\omega^2 r^2}{2g} = \frac{v_1{}^2}{2g}$$

where v_1 is the tip speed.

If, therefore, we could have the two causes of the rise in pressure in the delivery pipe in operation at the same time to the fullest extent, and if the air is thrown off from the tip at the same speed as that at which the tip itself is moving, we should have a total pressure H_1 produced by the fan as follows:—

$$\frac{v_1{}^2}{2g} + \frac{v_1{}^2}{2g} = \frac{v_1{}^2}{g} = H_1$$

or in other words, the pressure would be that of a column equal to twice the head $\frac{v_1{}^2}{2g}$ due to the tip speed.

Put in another way, the velocity in the air which might be generated by such a head would be $\sqrt{2}$ times the actual velocity of the tip of the vanes.

Unfortunately, however, it is impossible to have both these causes of increased pressure in full operation simultaneously, for if there is any flow of air through the fan it is obvious that the individual particles of air within the wheel will not follow the circular path round the centre of the axis, and therefore the calculation for increase of pressure, due to centrifugal force, is rendered void. Further, it is impossible to utilise the whole of the energy in the moving air by reducing its velocity to zero. There are also other losses of energy owing to shock and friction, and other causes to be shortly mentioned, which have not been taken account of.

The formula, however, gives what is called the "total pressure" of an ideal fan, and represents the extreme limit of the pressure which can be conceivably produced by the rotation of a fan of this type.

In practice the normal pressure which a fan can produce is about half this value, although the operation of most fans in certain conditions produces a pressure somewhat greater than the half of the theoretical maximum.

It will be observed that the velocity v_1 reckoned on in arriving at the above expression, is the velocity at the tip of the vanes, and in order to obtain the value for the ideal pressure

$$H_1 = \frac{v_1^2}{g}$$

it has been reckoned that the air is delivered at the tip velocity v_1.

Now the actual head H is always less than the ideal value H_1. In order to estimate the efficiency of a fan, an imaginary air velocity v is assumed, corresponding to the pressure H actually obtained. This velocity bears the same relation to the actual value H as the tip velocity v_1 does to the ideal value H_1, that is to say,

$$v^2 = \sqrt{gH}$$

Equivalent Orifice.

In order to compare fans of different dimensions and types on the same basis, it is customary to imagine that all the air is delivered at this velocity v from the fan through an opening in a flat plate of such a size as to represent the capacity of the fan. This opening, if a uniform stream of air at a velocity $v = \sqrt{gH}$ were maintained through it without any contraction, would pass the quantity of air Q actually delivered by the fan. This ideal imaginary opening is called the "equivalent orifice." Its area A is such that

$$Q = A\,v = A\,\sqrt{gH}$$

where H is the actual head.

In other words

$$A = \frac{Q}{\sqrt{gH}}$$

That this imaginary opening roughly corresponds to an actual opening which would pass the quantity Q of air, will be evident from the following :—

The actual velocity generated through any opening by a pressure H would be $\sqrt{2\ gH}$. But, owing to the contraction of the stream, as explained on page 65, the actual quantity of air passed would only be

$$Q = 0.65 \times a \times \sqrt{2\ gH} = 0.92\ a\ \sqrt{gH}$$

so that the actual opening differs from the ideal or " equivalent orifice " by about 8% only.

Manometric Efficiency.

What has been called the " manometric efficiency " of the fan is the ratio of the actual value of H to the ideal value H_1, in other words, of the square of the imaginary velocity v to the square of the actual circumferential velocity or tip speed v_1. That is

$$M = \frac{H}{H_1} = \frac{v^2}{v_1^2} = \frac{gH}{v_1^2}$$

Mechanical Efficiency of a Centrifugal Fan.

The most important problem in connection with fans from a practical point of view is to ascertain the mechanical efficiency. This efficiency may be estimated in several different ways, according to the purposes for which the fan is required.

In its most general sense the "mechanical efficiency" of any appliance may be described as the ratio of the useful work done to the work consumed by the appliance. In the case of the fan, the meaning of the expression "mechanical efficiency" depends on what is meant by "the useful work done by the fan," and also by "the work consumed by the fan."

(1). Useful Work done by the Fan.

If the fan were required to propel air at a high velocity, the efficiency might fairly be reckoned as the ratio of the work theoretically required to propel the quantity of air at that velocity to the work actually required to drive the fan spindle. On the other hand, if, as usual with a centrifugal fan, the object of the fan is to propel a volume of air against a certain definite pressure, the useful work would be fairly reckoned as that done against the pressure. The magnitude of both these two quantities is discussed in the earlier part of the present chapter. In so far as the fan is only required to produce pressure, the velocity of the air is a waste product of the fan, but it has been pointed out that by suitable design, the velocity at which the air is delivered may be made to add to the pressure produced by the fan.

It will thus be seen that the correct meaning of the "efficiency" in any given case depends not only on the fan itself, but also on the flues connected with it.

(2). The Work Consumed by the Fan.

There is an element of uncertainty also in regard to this point. The work required to rotate the spindle of an actual fan consists of two parts :—

(1) That necessary to rotate the fan runner itself, including that required to accelerate the air, and to overcome the friction of the air, as well as other air losses within the blades.

(2) The friction of the bearings of the spindle.

In practical cases the friction of the bearings has to be overcome in addition to the friction and inertia of the air. The friction of the journals can be reduced very largely by the adoption of ball or roller bearings without otherwise altering the design of the fan. The power necessary to overcome this bearing friction is not strictly a part of the work consumed by the fan as a fan, although in all practical cases it must be taken account of.

There is yet a further complication. In practice the fan, bearings and all, are invariably driven by an electric motor or some form of prime mover, which may be direct coupled to the fan, or it may be connected by means of a belt, or in other ways. The employment of a belt itself causes some loss. If the motor is electrically driven still further losses are caused both mechanically in the friction in the bearings of the motor and electrically within the motor itself, so that in all cases the amount of power which the proprietor of the fan has to pay for in order to keep it working is largely in excess of the power which is actually required on the fan blades in order to overcome the air losses.

From a scientific point of view the efficiency of the fan, considered as a pneumatic device and not as a piece of machinery, would be strictly the ratio of the useful work done (however that is calculated) to the amount of energy required to work the fan runner itself, apart from its bearings, and apart from belt or motor losses, but such an efficiency, if calculated, would be deprived of much of its practical utility on account of the uncertainty as to the amount of power which would have to be supplied to overcome the friction of the bearings.

All practical tests, therefore, are taken on the amount of power necessary to be supplied to the pulley of the fan, which includes the friction of the bearings and in some cases also belt losses.

This, though not exactly fair to the designer of the fan, is perhaps the most practical method of estimating the efficiency, because the customer can then equate the amount of power required for driving the fan to the amount of brake horse-power which will be given off from the pulley of the engine or prime mover, and if these two quantities are the same he knows that the engine is sufficiently powerful to drive the fan.

The ratio which is commonly recognised as the mechanical efficiency of the fan is for these reasons the ratio between the power theoretically necessary to propel Q feet of air per minute against the desired pressure to the power which must be given off from the pulley of the motor driving it.

The full expression for this efficiency is given on page 202, namely :—

$$\frac{Qh \times .000158}{\text{the horse-power of the motor.}}$$

Dynamic or Total Efficiency.

This air, however, is discharged at a considerable velocity, and it depends on the construction of the volute and the flues outside the fan how much of this velocity is converted into pressure. If the fan is well designed and provided with a trumpet-shaped outlet, or "chimney" as it is called, the pressure obtained can be considerably increased by a gradual reduction of the velocity. Assuming then that the air is discharged from the fan blades at a velocity v_1, and reduced by a carefully designed volute and outlet to v_2, it is shown on page 79 that it will be increased in pressure by $\frac{v_1^2 - v_2^2}{2g}$ feet of air column.

The useful work done by the fan is therefore increased by

$$Q \times .0761 \left(\frac{v_1^2 - v_2^2}{2g}\right)$$

It is obvious, therefore, that the efficiency can be reckoned in two ways.

(1) If the total work done by the fan is calculated on the pressure only, apart from the velocity at the fan outlet, the latter being calculated as wasted, the "pressure efficiency," as it is called, is the ratio of the work done per pound of air, or per minute in forcing the air against the pressure only, neglecting the velocity at which it is delivered, and the work which is necessary to supply to the shaft per pound or per minute in order to produce the result, or in other words, "the mechanical pressure efficiency" of the fan only (excluding the engine) equals

$$\frac{5.2\,Qh}{\text{Brake horse-power} \times 33{,}000.}$$

(2) On the other hand, the "total efficiency" has the following value :—

$$\frac{5.2\,Qh + Q \times 0.0761 \left(\frac{v_1^2 - v_2^2}{2g}\right)}{\text{Brake horse-power} \times 33{,}000.}$$

In both cases Q = cubic feet delivered per minute at 60° F.

The brake horse-power here means the power actually applied to the shaft of the fan, which is not necessarily the same as that given off on the pulley of the motor.

It must be clearly understood that a high total efficiency is only valuable when the kinetic energy generated can be actually used. It is no good to generate a good dynamic pressure merely for the purpose of wasting it, in order to get a good nominal efficiency out of the fan. One might just as well waste the kinetic energy at once without the trouble of converting it into pressure energy first.

It is thus clear that in considering the efficiency of a fan installation, the kind of duty the fan is doing must be considered as well as the nominal efficiency of the fan itself.

Having this principle in mind it would seem at first sight that where a fan is merely drawing air from a flue and discharging it into the atmosphere it is of no value to apply a chimney to the outlet, for a chimney merely has the effect of raising the pressure in front of the stream, and it would seem at first sight that this would be a positive disadvantage in the case of a fan drawing air by suction from, for instance, a mine.

This reasoning, however, is wholly wrong, as may be seen by considering the following :—

When the chimney discharges into the atmosphere the pressure at the *outlet* from the chimney must obviously be the pressure of the atmosphere, but owing to the action of the chimney this is greater than the pressure at the fan outlet. Therefore the pressure at the fan outlet must be *below atmospheric pressure*. Much more is the pressure at the eye of the fan below the atmospheric pressure.

In this case, the fan is really working in an atmosphere below the atmospheric pressure, and to discharge the air from that pressure into the atmospheric pressure is precisely the same thing as withdrawing air from the atmosphere and discharging it against a higher pressure still, into a flue. So that a chimney is, in this case, just as useful as it is in the case of a flue.

The function of a suction fan is in fact to take air from a pressure lower than the atmosphere and raise it to the atmospheric pressure. The difference between the operation of a suction fan and a propelling fan is therefore one of degree only.

Volumetric Efficiency.

The various kinds of efficiency described above, though no doubt scientific, are somewhat too involved to be of immediate use to the designer of fans for giving him an idea of the size of the fan required for any given duty.

There has therefore been another expression devised, known as the "volumetric efficiency" of the fan, which is designed for the strictly practical object of giving an immediate idea of the size of runner required for any given duty.

A square flue is imagined, whose side is equal to the radius of the fan wheel. Air is supposed to be propelled along this flue with the same velocity as that of the tip of the blades.

The volumetric efficiency is the ratio between the quantity of air actually delivered by the fan and the quantity which would be delivered by the flue, and at the velocity just described, or

$$\delta = \frac{Q}{v_1\ r^2}$$

If, therefore, the designer knows the "volumetric efficiency" of any given type of fan, he can at once determine what the diameter of the wheel will be at a given number of revolutions.

Fans of the same type and make are generally made with a constant relation or ratio between their breadth and the diameter, so that knowing the volumetric efficiency of the particular design, an approximate determination of the diameter is soon made.

It must be clearly understood that the "volumetric efficiency" is merely a rule of thumb, and has no definite relationship with the scientific efficiency of the fan. Its value may be, and often is, above 100%, showing that the term "efficiency" is wrongly applied to such a function as this. Also the volumetric "efficiency," so described, takes no account of the width of the blades.

Now the actual delivery of a fan against a constant pressure (but not, of course, against a constant resistance) is almost proportional to the width of the blades, so that any formula professing to represent the volume discharged which takes no account of the width is obviously wrong.

A better expression for the volumetric efficiency would be

$$\frac{Q}{v_1\ D\ B\ \pi}$$

The two values would be the same in cases where

$$\frac{r}{2} = \pi\ B$$

but even the latter expression could not represent "efficiency" in any scientific sense. All such expressions are merely rules of thumb. The latter is more based upon common sense than the former.

Power Exerted on Fan Wheel.

Now consider the amount of power which must be exerted by the fan wheel on the air in order to produce these results.

If momentum is acquired by any mass acted upon by any force, then the product (called the "impulse") of the force in poundals into the time during which it acts is equal to the amount of momentum acquired by the body in that time.

Further, the product of the "impulse" into the perpendicular distance from any point in its plane of motion is equal to the amount of momentum acquired by the body during the same interval multiplied by the perpendicular distance of its line of action from the same point.

A further development of the same theory is that if any *change* in the momentum of a body is due to the action of a force which is variable, both in direction and magnitude, then the difference between the ultimate momentum of the body multiplied by its distance from any point in its plane of motion, and the original momentum also multiplied by the distance from the same point, is equal to the sum of the products of the mean forces acting during each instant, each multiplied by the time through which that force acts, and each multiplied by its vertical distance from the same point as before.

A concise expression of this law is as follows:—

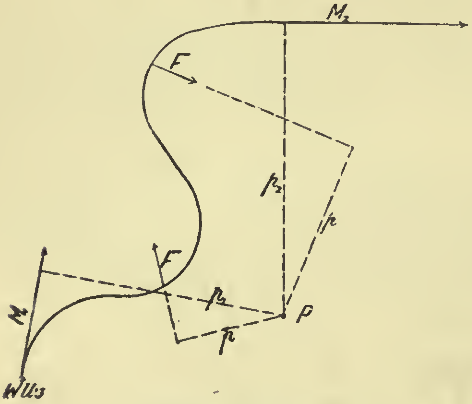


FIG. 58.

That the change of moment of the momentum of a mass acted on by any variable force during any given time is equal to the moment of the impulse of that force during the same time.

Thus, suppose that in the figure a body of mass W pounds has travelled along the path indicated on the diagram, Fig. 58, during a certain time, which, for the purposes of fixing our ideas, we will assume is 0.1 second, and that it is constrained to move along this path by the action of a variable force F, which is indicated by an arrow in several different positions, and as a result of the action of the force the momentum has changed from M_1 to M_2.

Now assume that the total time during which the force acted on the mass, namely, 0.1 second, is divided into a hundred equal parts, each equal to .001 of a second, and that the average value and direction of the force F acting during each of these small intervals is determined in poundals, then our theory states that

$$(M_2 \times p_2) - (M_1 \times p_1) = \Sigma\,(F \times .001 \times p)$$

Now this law is necessarily and exactly true for every particle of air passing through the fan.

It is clear that as the action of the fan is continuous the moment of all the forces acting on all the particles of air in the fan at one instant about the centre of the spindle is constant, and that this is the twisting moment on the fan shaft.

Now let W be the weight of air moving through the fan per second, and v the component of the velocity tangential to the wheel with which it leaves the fan blades. Then the moment of momentum of the air leaving the fan during any short interval $\triangle t$ seconds must obviously be the same as the moment of the impulse of the forces acting during that same short interval on every particle of air within the fan blades, although the result of the action of this latter will only be manifested during later intervals $\triangle t$ in the continuous discharge of the same quantity of air per interval, as during the interval in question.

Now the moment of the momentum of the air leaving the blades during the interval $\triangle t$ is obviously

$$\triangle t \times W \times v \times R$$

and the sum of the moment of the impulses of the forces acting on every particle of air in the wheel during the same interval must obviously be

$$T \triangle t$$

Hence

$$W v R = T \text{ in poundals-feet,}$$

or in pounds-feet

$$\frac{W v R}{g} = T$$

Multiply each side of this equation by ω and we have

$$\frac{W v R \omega}{g} = T \omega =$$

work done on fan shaft in one second, from which the tangential velocity of the air leaving the blades can be calculated.

But as $R \omega = v_1$ the tip velocity, we have

$$\frac{W v v_1}{g} = \text{the work done in one second} = T \omega$$

and the work done on one pound of air obviously

$$\frac{v v_1}{g}$$

Stated in words, this important result is as follows :—

The moment of the momentum of the whole mass of air leaving the fan in one second is equal to the twisting moment on the fan spindle, multiplied by the angular velocity of the wheel, excluding, of course, the frictional resistance of the bearings of the fan.

From this we can derive a useful expression for the pressure efficiency of the fan.

Useful work done per pound of air $= H$.

Work done by fan shaft per pound of air $= \dfrac{v v_1}{g}$

Hence :—

$$\text{Pressure efficiency} = \frac{g\,H}{v\,v_1}$$

Effect on the Velocity of the Shape of the Vanes.

It is obvious that the shape of the vanes must have considerable effect, not only on the loss of head in the air as it passes through the fan, but also on the efficiency of the fan. If the vanes are so shaped that they discharge the air passing through the fan at a high velocity, it is obvious that the fan wheel will absorb more power than if it were discharged at a low velocity.

If the velocity at which the air is discharged is wasted, owing to the conditions of the casing and flue surrounding the wheel, it is sheer waste of power to generate a high velocity in the outflowing air.

If, on the other hand, arrangements are made for utilising the velocity of the outflowing air and converting it into pressure, and thus into useful work, it will certainly increase the capacity of the fan, and it may be the greatest possible economy to discharge the air at a high velocity.

In order to understand the bearing of this question on the design of a fan, it will be well to consider the way in which the air actually does pass through the fan.

It is obvious to begin with, that the quantity of air passing into the eye of the fan (as the part at the centre of the vanes surrounding the shaft is called) must be identical with that which is thrown off from the tips.

If r_1, b_1, are the radius and breadth of the fan at the interior of the blades, and r_2, b_2, on the outside circumference of the vanes, and u_1, u_2, the corresponding components of the velocities of the air calculated in a radial direction, it is clear that

$$Q = 2\pi \times r_1 \times b_1 \times u_1 = 2\pi \times r_2 \times b_2 \times u_2$$

or

$$u_1 = \frac{Q}{2\pi\,r_1\,b_1} \text{ and } u_2 = \frac{Q}{2\pi\,r_2\,b_2}$$

The total velocity of the air delivered is compounded of this radial velocity, and that tangential to the wheel, where v_t is the tangential velocity, and u_2 the radial velocity.

The value of the resultant velocity is obviously

$$\sqrt{v_t^2 + u_2^2}$$

Now the proportion between v_t and v_1, the velocity of the wheel itself, depends on the shape of the tip of the vane. If the vane is bent backwards, so that it strokes the air without carrying it along, it does not give the air such a large tangential velocity as it does when the tip of the vane is radial. This again does not give the air such a large velocity as it does when the tip of the vane is bent forwards. These differences will be understood from an examination of Fig. 59.

The tangential velocity of the air may indeed be considerably greater than the circumferential velocity of the wheel when the tip of the vane is bent forwards.

It is therefore clear that when the air is merely to be discharged into the open atmosphere without a casing, and when therefore the velocity energy is all to be lost, the vanes should be bent backwards, so as to stroke the air into the atmosphere at as low a velocity as possible.

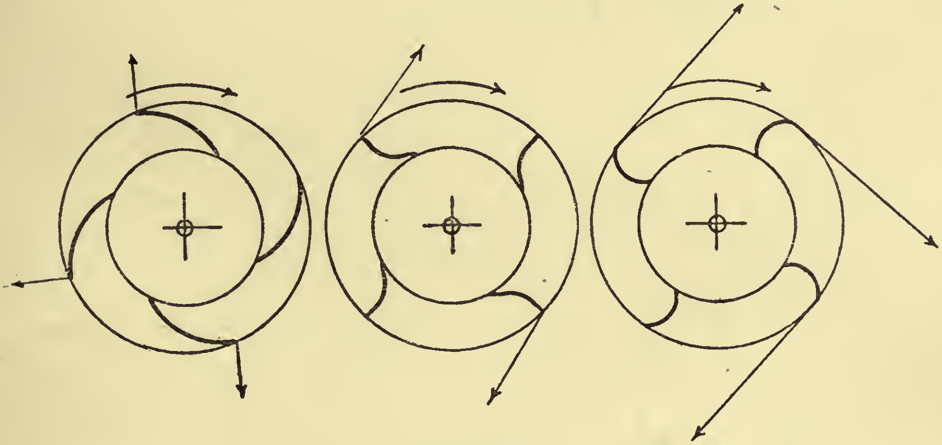


FIG. 59.

On the other hand, when it is desired to attain a very high pressure as in blowing fans, and when provision is made for utilising the velocity by a gradual enlargement of the passage, it is desirable that the velocity of discharge shall be as great as possible. In this case the tips of the vanes are radial or bent forwards, in order to give a sting or fillip to the air as it leaves the fan blades.

Now the efficiency of a fan depends on a large number of other conditions, and it is impossible to deal with the matter fully except in a treatise especially devoted to the fan. The ventilating engineer has mainly to do with fans discharging air against a comparatively low water gauge, generally less than 1″ and never greater than 2″ to 3″ at the outside. The theory of the fan, unless it is treated in a highly complicated manner, cannot possibly take account of all the practical conditions on which the efficiency depends. Therefore, for the purposes of the ventilating engineer, it is only possible to deal with the matter from the experimental standpoint.

Losses in a Fan.

Without going thoroughly into the theory it will be well to indicate the causes of loss of pressure in a fan, which give rise to the differences which exist between the theoretically perfect fan and the practical one.

(1) To begin with, air entering the eye of a fan of ordinary construction moves at first in a plane which passes through the axis of the fan. Neglecting the component of this velocity in an axial direction, and considering only the radial component, as soon as any particle of air is caught by the inner edge of the blade, a sudden change of velocity is communicated to it. The suddenness of the change depends on the angle which a central section through the blade makes with a plane tangential to a cylinder co-axial with the spindle at the inner edge of the blade. If this angle is small, as it should be, the shock may be diminished, and for one particular delivery of the fan entirely avoided. This is only true for one particular delivery, that is one particular radial velocity of the air. For any other velocity there must be of necessity some shock due to sudden change of direction.

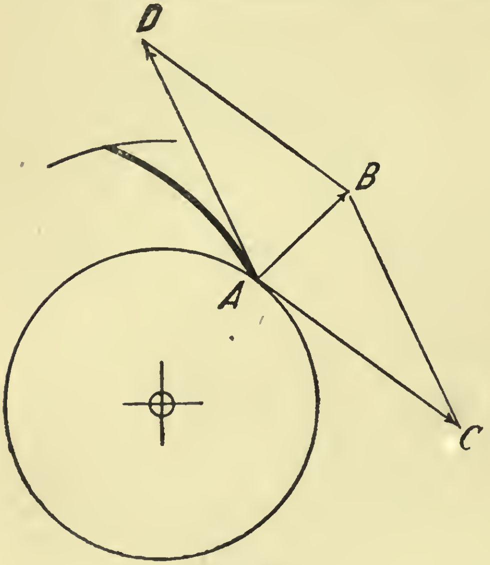


FIG. 60.

Thus, if air is moving radially with a velocity *A B*, the inner edge of the fan at the instant in question with velocity *A C*, the relative velocity between a particle of air *A* and the fan blade will be *A D*. If the fan blade is made tangential to *A D* the air will slide up the blade of the fan without any shock at all.

(2) The second cause of loss of head is friction within the blades of the fan itself. This can be minimised by making the blades with very smooth curves, and having as few projecting bolts and rivets and other irregularities or roughnesses of the surface as possible.

(3) The next cause of loss is the sudden change of velocity on passing from the fan itself into the volute, or in cases where there is a diffuser, from the diffuser to the volute. There is no loss of energy in passing from the fan into the diffuser, for the diffuser is virtually part of the fan wheel, in which there are no blades.

(4) The next cause of loss is friction in the volute.

(5) Finally, any sudden changes of velocity between the volute and the chimney, or between the volute and the flues.

The magnitude of these losses can be calculated approximately from theoretical considerations only, but the calculation involves more mathematics than it is proposed to go into in the present work. For practical purposes the total losses can best be measured experimentally, by determining the efficiency of fans of various types.

General Observations on the Power Employed in Working Centrifugal Fans.

It will be understood from the preceding that as the function of a centrifugal fan is to deliver a certain quantity of air at a given pressure, the criteria of excellence are the capacity to perform the duty at a minimum speed, with a minimum consumption of power and with a minimum production of noise and occupation of space.

The exact power conditions under which a centrifugal fan works are somewhat difficult to grasp, owing to the simultaneous variation of several variables. In order to make the relations between these variables clear it will be well to consider the simultaneous variations of two of them at a time.

Delivery and Power with Fan Running at a Constant Speed.

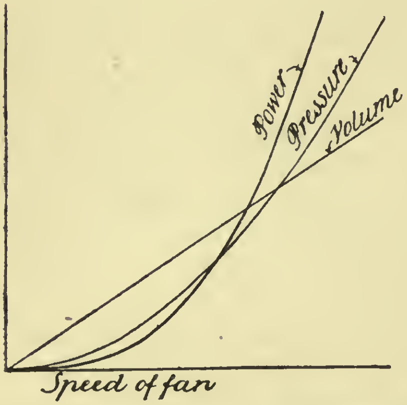
Suppose a centrifugal fan running at a constant speed discharges air into a chamber from which there is an outlet, the size of which can be adjusted by means of a damper. When the damper is closed the rotation of the fan at this speed (as already explained) will produce a certain pressure, due only to the centrifugal force which the rotation of the air around the axis generates. The magnitude depends, as has also been shown, only on the tip speed of the fan. If that speed is increased the pressure will also be increased proportionally to the square of the number of revolutions per second.

There is no air passing away from the delivery and therefore no air is passing through the fan. In these circumstances the power absorbed by the fan will be at a minimum. It will be merely what is required to overcome the friction. The mere maintenance of the pressure absorbs very little power. No air is being accelerated and the power is not absorbed in that way. There is no air given off at a high velocity at the tip of the fan, and this therefore absorbs no power.

The fan wheel and the air within it may then be imagined as forming a sort of fluid fly wheel enclosed in a casing.

When, however, the outlet from the chamber is slightly opened, air begins to pass through the fan. The power absorbed increases because work is now being done in accelerating a certain weight of air every second from zero velocity up to some velocity probably in the neighbourhood of that of the fan tip, but depending on the exact shape of the blades. The wider the damper is opened the greater is the amount of air and the greater the absorption of power.

The gradual opening of the damper also produces another effect. When the damper is absolutely closed, the pressure generated is simply that due to the centrifugal force. But when the damper is partly open this pressure may and does in some cases actually increase, because of the additional pressure produced by the velocity with which the air is discharged from the fan. The magnitude of the increase depends, of course, on the design of the volute and chimney.



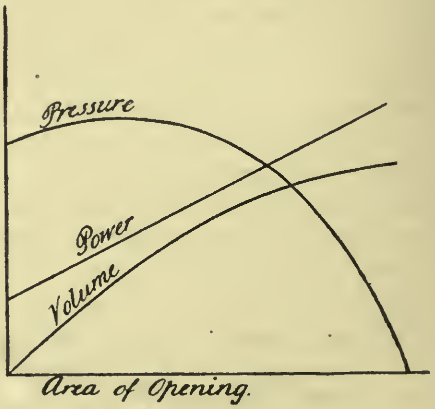


Diagram illustrating variations
in delivery, pressure and power

with increasing speed when
Resistance is constant

FIG. 61.

with increasing opening when
Speed is constant

FIG. 62.

According to the principles already explained, if these are designed so as gradually to reduce the velocity, an additional pressure is generated over and above that produced by the centrifugal force.

As the opening becomes wider and wider, the amount of air increases, and so in consequence does the power required to drive the fan. The pressure after first rising slightly gradually begins to fall, and continues to do so until when the fan outlet is freely open to the air, the air discharged and the power absorbed are at a maximum, and the pressure can be no higher than the atmospheric pressure. The variations in the pressure, etc., in these circumstances will be understood from an inspection of Fig. 62, in which all these simultaneous variations are plotted on a base representing the amount of damper opening. These may be called the characteristic curves of the fan.

Speed, Volume, Pressure and Power Against a Constant Resistance.

Consider now the effect of gradually increasing the speed of the fan while maintaining the resistance (that is the damper opening) constant. As the speed rises it is clear from what has been already explained, that the total pressure produced by the fan must increase in proportion to the square of the speed, in other words, the speed must increase proportionately to the square root of the pressure.

It has been shown in Chapter V. that the amount of air which a given pressure will force past a fixed resistance also increases as the square root of the pressure. Since both the speed of the fan and the velocity of the air produced in the flue are therefore proportional to the square root of the pressure, it is evident that the amount of air delivered must be proportional to the speed of the fan. The pressure is proportional to the square of that speed; the horse-power absorbed, as has been shown on page 202, is proportional to

$$Q\ (5.2\ h + .00118\ v^2) \text{ foot pounds.}$$

As Q is proportional to the speed, and h and v^2 are both proportional to the square of the speed, it is clear that the whole expression must be proportional to the cube of the speed.

Plotting therefore the delivery, the pressure, and the power on a base representing the speed of the fan, as in Fig. 61, the relation of these various quantities will be understood.

In order to illustrate the same point a series of characteristic curves are given for the same fan when operated at different speeds, the speed being constant for each curve, and the delivery being varied from zero to a maximum at each speed. A comparison of these is very instructive.

The rules, therefore, for a fan working *against constant resistance* are :—

The air delivered varies as the speed.
The pressure varies as the square of the speed.
The horse-power varies as the cube of the speed.

Efficiency.

Consider the variations of the mechanical efficiency when calculated for pressure alone. When the damper is shut no air is being delivered and in consequence no useful work is being done, although considerable power must be employed in working the fan. The efficiency is therefore zero in these conditions. On the other hand, when the damper is wide open, though a large volume of air is being delivered, it is discharged against no pressure. The value of $Qh \times .000158$, which represents the pressure horse-power in the air, is again zero. The efficiency in these circumstances also must be zero. *Somewhere* between these two points the efficiency must rise to a maximum value. The curve of efficiency thus has the shape shown on Figs. 63 to 70.

When the kinetic energy of the outlet air is taken into the calculation as useful energy, the total efficiency is clearly zero when the damper is shut, but unlike the mechanical pressure efficiency it may rise continuously with the delivery, or more generally, it may drop slightly when the pressure is nearly atmospheric, but in any case the total efficiency must always be greater than the mechanical pressure efficiency.

A centrifugal fan is, or should be, always required to work against pressure. If it is merely required to move air against little or no pressure a centrifugal fan should not be employed at all. It is waste of power and money to employ a centrifugal fan for such a duty as this. Although, therefore, it is scientifically more satisfactory to calculate the mechanical efficiency on a total efficiency basis, that is to say, as a ratio of the visible energy produced to the visible energy employed, there is practical reason for estimating the value of the efficiency on a pressure basis alone, although the value of the pressure efficiency does not depend on the fan alone, but also on the arrangement of flues adopted with it.

Assuming that the power absorbed bears a fixed ratio to the visible work done, that is to say, assuming that the total efficiency of the fan remains constant between given limits of delivery, which the diagrams show it does approximately, it is clear that excluding friction the power actually absorbed must be proportional to the following :—

$$Q\ (5.2\ h\ +\ 0.00118\ v^2)$$

The power absorbed can, of course, only be determined experimentally. It will be seen from the diagram that in these circumstances the power increases with approximate regularity with the delivery from a certain minimum at no output up to a maximum at free discharge. This relation, of course, fixes an empyrical relation between h and v according to the above equation.

It is important to observe that the previously quoted rule, namely, that the power varies as the cube of the speed applies only to the case

TEST OF 4′ 6″ DIA. FAN AT VARIOUS SPEEDS.

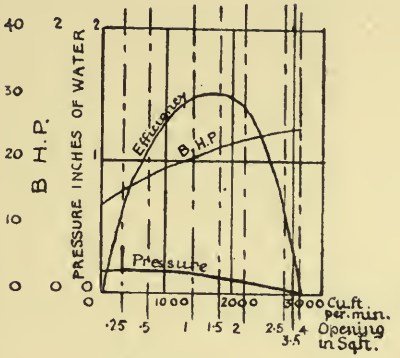


FIG. 63.

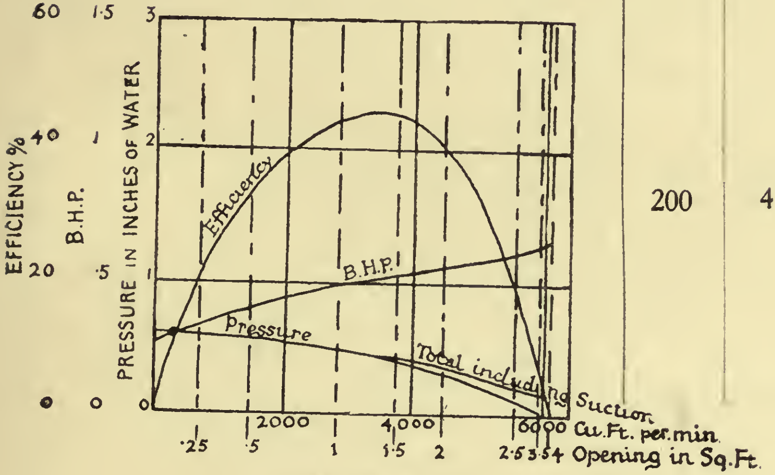


FIG. 64.

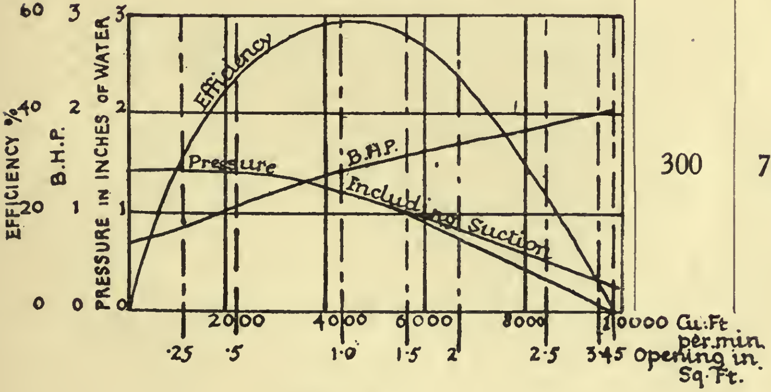


FIG. 65.

R.P.M.	Tip Speed v ft./sec.	Value of $\frac{v^2}{g} \times \frac{12\,D_a}{D_w}$ = ideal pressure in ins. water.
100	23.5	0.25
200	47	1.00
300	708	2.28

TEST OF 4′ 6″ DIA. FAN AT VARIOUS SPEEDS.

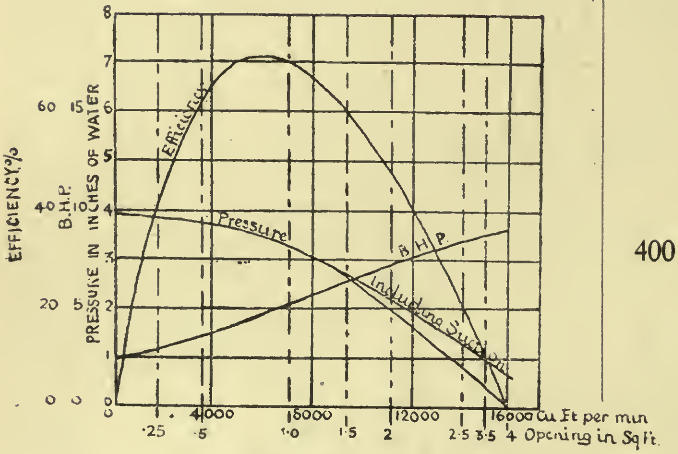


FIG. 66.

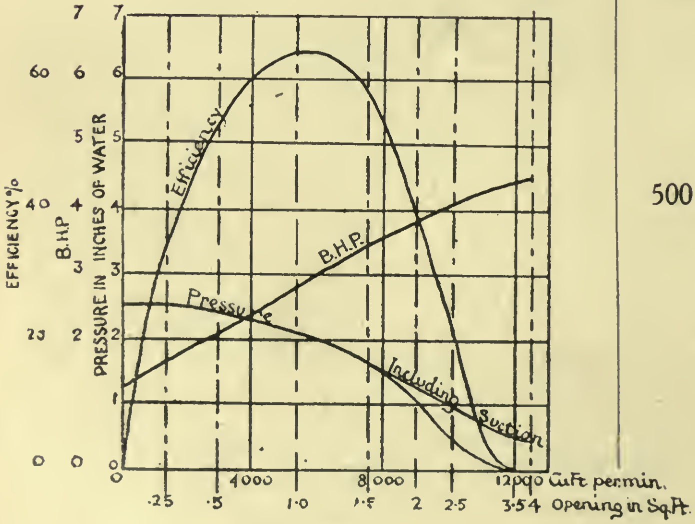


FIG. 67.

R.P.M	Tip speed v ft./sec.	Value of $\frac{v^2}{g} \times \frac{12\ D_a}{D_w}$ = idea pressure in ins. water.
400	94	3.97
500	118	6.4

TEST OF 4′ 6″ DIA. FAN AT VARIOUS SPEEDS.

R.P.M.	Tip speed v ft./sec.	Value of $\frac{v^2}{g} \times \frac{12\,D_a}{D_w}$ = ideal pressure in ins. water.
600	141	8.8
700	164	12.2
800	188	16.0

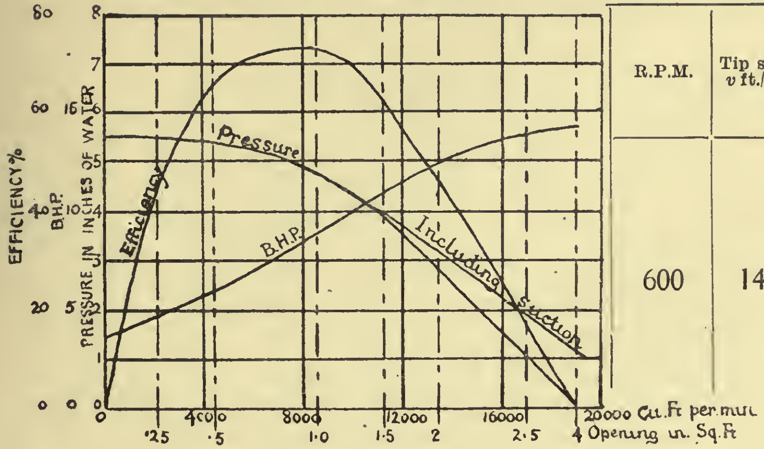


FIG. 68.

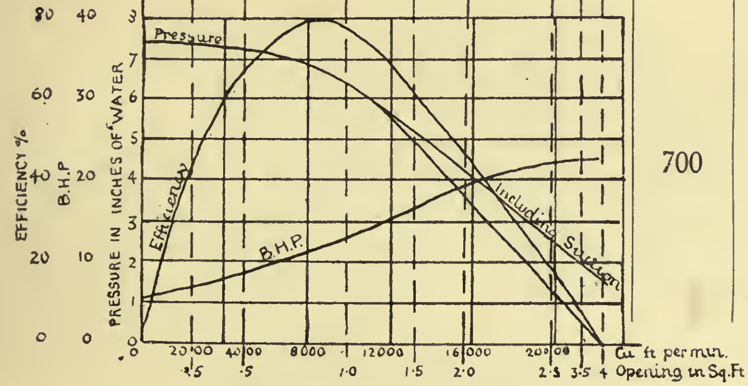


FIG. 69.

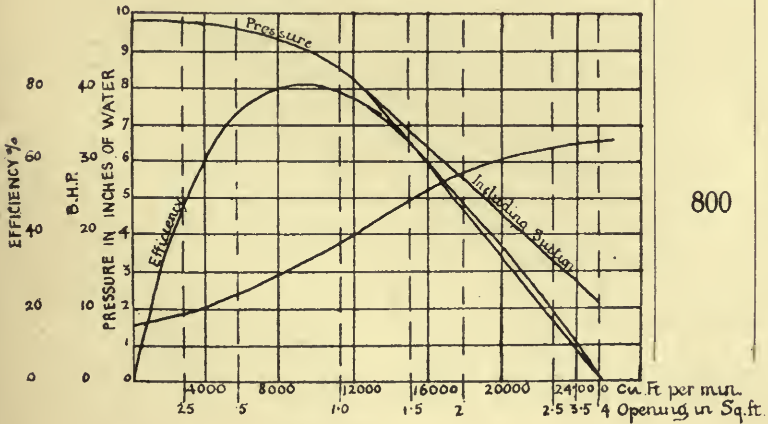


FIG. 70.

where the fan is working against a constant resistance. In this case the resistance is not constant and therefore the three fundamental rules do not apply.

The following are the rules applicable to this case :—

At a constant speed the power absorbed =

$$M\ Q + C$$

where M and C are constant. The value of C is proportional to the speed and the value of M proportional to the square of the speed. The total efficiency is, of course, zero at no delivery and increases to a maximum when the delivery is about half the volume delivered at free discharge.

At this point the pressure efficiency, the compression, and the total gauge are approximately at a maximum, and this is generally about the point at which all centrifugal fans should be worked.

If the fan blades are suitably designed the pressure at this delivery is approximately at a maximum. The actual pressure in feet of air column at this delivery may be found from the formula

$$H = \frac{M\ v_1^2}{g}$$

where M is what has been called the manometric efficiency.

Figs. 63 to 70 represent the results of a comprehensive test of a fan with a runner 4′ 6″ diameter, by running it at speeds varying from 100 to 800 R.P.M. At each speed the delivery was varied by a damper from zero to a maximum and the corresponding deliveries, pressure power and efficiency noted. These are plotted on the curves and form a very complete illustration of the features of this particular fan and of the principle of fans in general.

General Remarks on Design of Flues.

From the above discussion it is evident that in designing a system of flues it is important that the pressure required to drive the air through them shall be as low as possible. In proportion, as the pressure necessary to drive a given quantity of air through a system of flues rises, the power theoretically required to force the air against the resistance into the flues increases in the same proportion, and the power necessary to generate the velocity in the air increases as the square of the velocity.

Any alteration of design, therefore, which will enable the required quantity of air to be delivered at a low pressure and low velocity is very desirable in the interests of economy.

In order to illustrate this point, consider an installation in which 500,000 cubic feet of air are to be delivered per hour into a building. A system of flues is designed which will require a pressure of, say, 1½″ of water to force the required quantity of air into the building. The peripheral speed of the fan will therefore be

$$\sqrt{\frac{1.5 \times 68.3 \times 32.2}{M}}$$

or assuming M to be 75%—an average good value—roughly, 66 feet per second. The horse-power required for the fan will probably be about

$$\frac{500,000}{60} \times 1.5 \times \frac{0.000158}{0.6} = 3.3 \text{ H.P.} \qquad \text{(see p. 202)}$$

assuming a fan of ordinary good construction, having a mechanical efficiency of, say, 60%.

Assume this horse-power is derived from electricity supplied at 1½d. per unit, and that the installation is working for 300 days in a year at 10 hours a day, the cost of electric current would therefore be about

$$\frac{3,000 \times 3.3 \times 1.5}{240}$$

£62 per annum.

If now the flues are doubled in area, with a consequent increase in the expense, the same amount of air could be delivered at half this velocity of the air, the necessary pressure would only be about 0.375″, the reduction in the necessary speed of the fan would be 50%, the reduction in power would be 75%, and the cost would be about, say, £17 per annum, or a saving of about £49 per annum.

The difference in the cost of the fan and flues might perhaps be £100. The results would be a gain of 50% on the additional capital outlay, or the additional cost could be paid in two years.

Comparisons of Efficiency of Similar Fans of Different Sizes Working on the Same Duty.

In comparing the performance of two fans of the same design and proportional in all their parts, it must be observed that the volume delivered in each case is proportional to the area of the outlet multiplied by the velocity at which the air is delivered, or in other words, proportional to the square of any one linear dimension multiplied by the velocity.

If in comparing two similar fans the output is to be the same, and assuming that the linear dimensions of the larger fan are q times that of the smaller, and assuming that d is the length of the side of the square outlet of the smaller fan, we must have

$$d \times d \times v_1 = d \times q \times d \times q \times v_2$$

or

$$d^2 \times v_1 = q^2 d^2 \times v_2$$

or

$$\frac{v_2}{v_1} = q^2$$

It is therefore clear that as the power absorbed is, *for a given discharge*, proportional to the square of the velocity of the air, this portion of the power absorbed or wasted in producing the velocity in the smaller fan must be q^4 times that absorbed in the same process in the larger.

It is thus clearly desirable that as large a fan as possible should be used in the interests of economy of power, so as to reduce as far as possible the waste of power involved in delivering the air at a high velocity.

If the two fans are run at the same tip speed, the *total pressure* produced may be assumed to be the same in the two cases. The work done actually against the pressure in delivering the same volume of air will be the same, but since with the smaller fan the velocity in the air must increase through the fan and through the opening in proportion to the square of one linear dimension, it is clear that this alone will cause the absorption of a good deal of extra power and therefore a diminished efficiency for the same output, and a considerable increase in tip speed in order to generate the additional necessary pressure for overcoming the enhanced resistances.

If the larger fan is working at its point of maximum efficiency in delivering this quantity of air, the smaller one will be working at a most unfavourable point at which the pressure produced will not correspond to the tip speed.

In determining then what size of fan to employ, it is necessary that the heating engineer should be informed what is the point of maximum efficiency. For ordinary designs of fan he can generally determine this point approximately for himself by ascertaining what is the delivery of the fan at the given speed with free intake and discharge. If he divides this quantity by 2 he can be sure that in general the fan will then be working not far from its maximum efficiency.

Example.

The ventilating engineer may apply the rules here given in the following manner. Assuming he has submitted to him a fan which is claimed to deliver 1,000,000 cubic feet of air per hour against a water gauge of ⅞″, the power absorbed being 4½ horse-power. The fan to run at 330 R.P.M., the runner being 35″ dia.

It is first necessary to calculate what mechanical efficiency this represents in the following manner :—

Mechanical efficiency =

$$\frac{Q\,h \times .000158}{\text{H.P. required}} = \frac{17{,}000 \times 0.875 \times .000158}{4.5} = 52\%$$

It will be seen from the published results of experiments previously alluded to that 70% is a good value for the mechanical efficiency, 50% is medium, and 30% is very poor. He should decline to accept any fan having a low mechanical efficiency in the interests of his client. The manometric efficiency may be calculated in the following manner :—

$$M = \frac{g\,H}{v^2}$$

Here $g = 32.2$.

$$H = \tfrac{7}{8}'' \times 68.3 = 60 \text{ feet air column.}$$

$$v = \frac{35'' \times \pi}{12} \times \frac{330}{60} = 50.3 \text{ feet per second.}$$

$$M = \frac{32.2 \times 60}{(50.3)^2} = 76\%.$$

The volumetric efficiency is not much of a guide since it is a mere rule of thumb which may or may not be useful as a guide to the determination of the dimensions of a fan of given design, but is of no value in comparing fans of different make.

In this case volumetric efficiency calculated on the ordinary basis

$$= \frac{278}{\left(\frac{17.5}{12}\right)^2 \times 50.3} = 261\%$$

The engineer should be careful to ascertain that (1) the results claimed will be guaranteed by the maker of the fan, (2) the speed at which the fan is normally to work should be the speed at which it has a maximum efficiency, and (3) that that efficiency is reasonably good. He should also compare the power necessary to drive the fan in question with fans of larger and smaller sizes. It is very desirable also that he should be provided with the characteristic curves of the fan running at the given speed so as to know what will be the effect of alterations in the resistance if his calculations of the flue resistances have been incorrect.

Experiments on Centrifugal Fans.

A valuable series of experiments, illustrating the points above explained, were made by Mr. Bryan Donkin and Messrs. Heenan & Gilbert, particulars of which are to be found in "Proceedings of the Institution of Civil Engineers," Vols. CXXII. and CXXIII.

Mr. Bryan Donkin tested a series of 11 different fans, about 10 experiments being made on each fan, each occupying about 15 minutes; the diameters of fans varied between 6″ and 25½″. The particulars of the experiments may be obtained from the paper.

From these experiments a very comprehensive idea of the efficiency of a large number of different types of fan may be obtained.

Heenan and Gilbert's experiments were made on one type of fan. Their object was to determine the best shape of fan blade and casing, and to obtain data whereby the proper diameter of such best type of fan and its most economical speed could be determined.

Propeller Fans.

The theory of the propeller fan has not been worked out so completely as that of the centrifugal type. Any one theoretical analysis would only be applicable to one special shape of blade. Such an investigation would be so excessively complicated as to be practically useless, except, perhaps, in the case of very simple surfaces such as plain flat blades.

Even in this case an experimental investigation is much more useful.

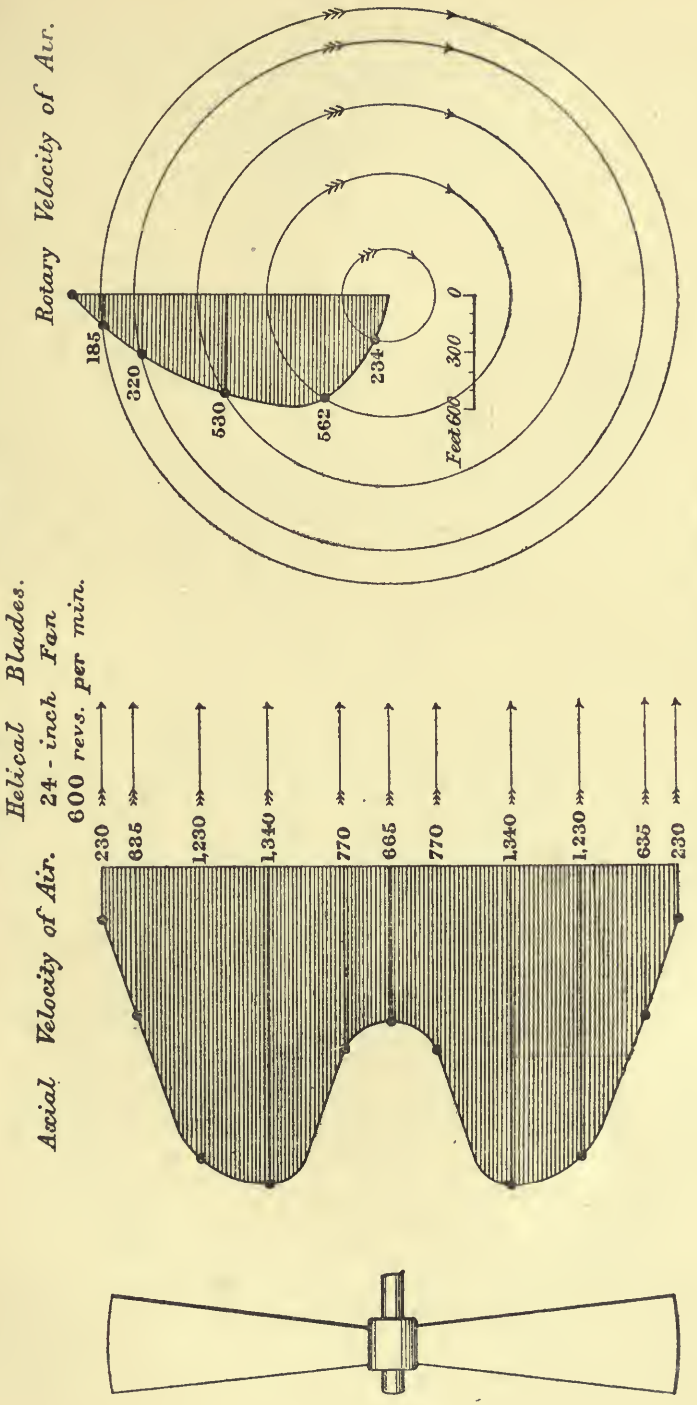
General Description.

The propeller fan is perhaps best regarded as working on the same principle as an ordinary multiple thread screw. The air propelled may be regarded as the nut, and the fan itself as a short portion of the screw. The nut cannot, however, be regarded as a plain cylinder having a hole through the centre as an ordinary iron nut. It is found experimentally that the air flows on the delivery side more or less as a complete cylinder, except for the space occupied by the shaft and boss of the fan. On the suction side, however, the air approaches the fan from all sides.

The working of a fan of this type produces its result in the following manner :—

The rotation causes the forward edge of the blade at the suction side of the propeller to cut into more or less stationary air and force it forward on the wedge or inclined plane principle. The inertia of the air forms the lateral resistance necessary for the operation of a wedge. The forward motion of the air propelled produces a partial vacuum in its neighbourhood behind the propeller. Air at the back of the fan rushes in to fill this vacuum. The momentum of this air carries it into the blades by which it is caught, pressed forwards, and discharged with a sort of spiral motion from the forward side of the rotating fan. The velocity of the issuing air has thus a considerable component in a direction parallel to the axis, and a component also in a circumferential direction. The magnitude of each component depends on the angle of the blade.

PROPELLER VENTILATING FANS.



Figures denote Velocity in feet per minute. Scale 1/8th.

Mechanical Engineers 1897.

FIG. 71.

It is clear that the circumferential component of the velocity is useless for the purpose for which the fan is generally required, which is to propel a given volume of air from one side of the fan to the other. The only motion which is useful for this purpose is the axial motion.

The current of air delivered, however, does not spread much beyond the outside diameter of the rotating vanes, in spite of the spiral or circumferential velocity which, of course, would of itself give the air a centrifugal tendency to spread. The air is delivered in fact almost as a solid cylinder in cases where the propeller is working against no resistance or counter-pressure. The reason for this, probably, is that if the moving air were to fly outwards it would leave a partial vacuum at the core of the cylinder. The external pressure of the atmosphere is sufficient to prevent the formation of any considerable vacuum. This vacuum, such as it is, is probably partly instrumental in drawing in air through the central part of the propeller which would otherwise pass no air. The external pressure therefore tends to keep the air delivered in the form of a solid cylinder, rotating as a whole in the same direction as the fan itself, in which both the longitudinal and circumferential components of the velocity near the axis are less than near the circumference.

There can, however, be no doubt that the absolute pressure of the air at the centre of the core must be to some slight extent lower than that at the outside of the cylinder of the propeller near the axis.

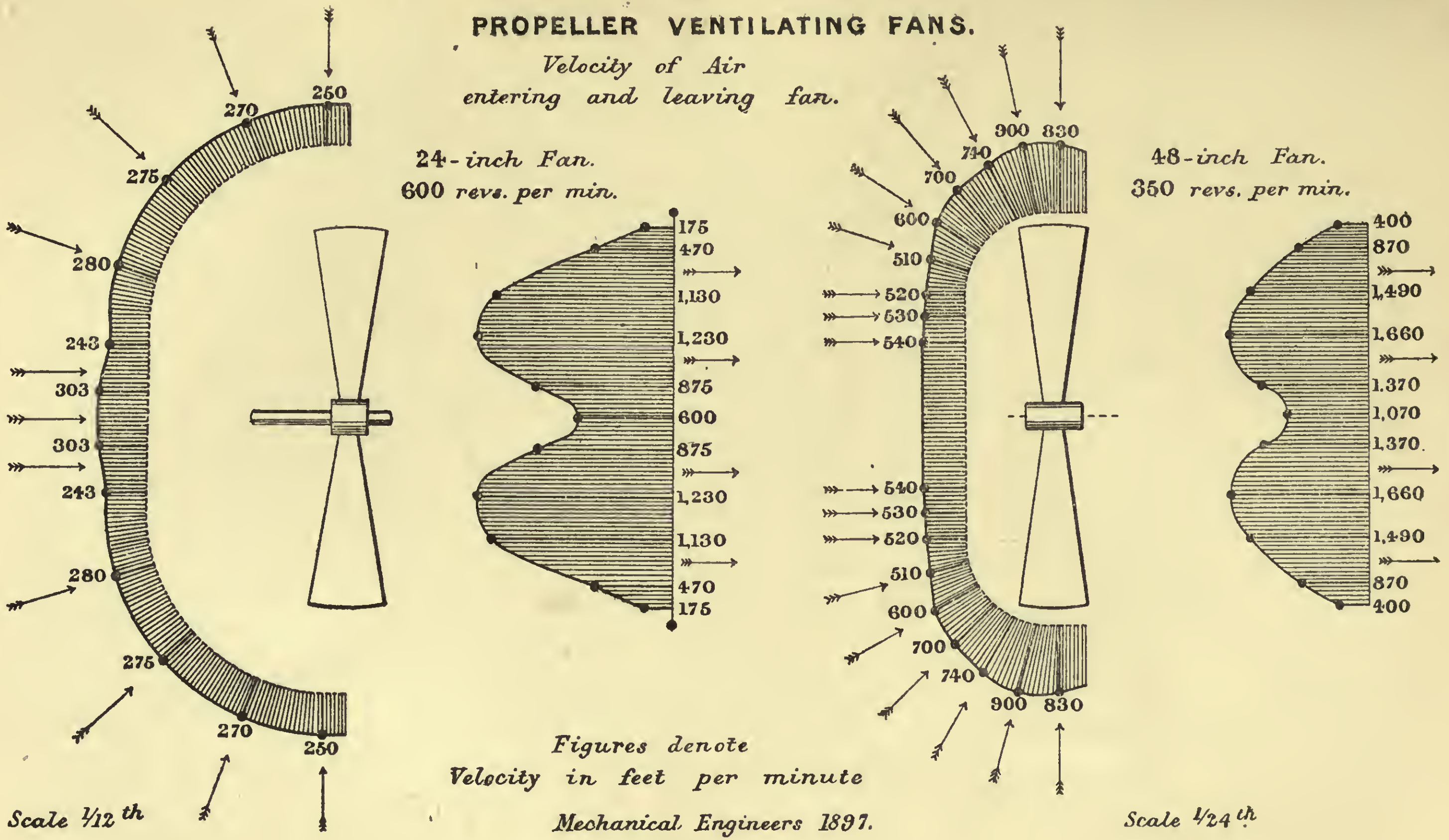
The working of a flat-bladed fan is clearly illustrated by Figs. 71 and 72, representing results of tests reported in a valuable paper by Mr. W. G. Walker, read before the Institution of Mechanical Engineers in 1897. The magnitude of the velocity is shown on these diagrams all round the fan, being presumably from considerations of symmetry the same at all points which are equally distant from the axis of rotation.

Fig. 71 shows the magnitude of the axial and circumferential components of the velocity of the delivered cylinder of air.

On the left of the fan, in each case, Fig. 72 shows that far from having any centrifugal effect, the fan draws air inwards around its periphery if placed in the open air unsurrounded by any tube or other obstruction, and that it discharges the air in an approximately axial direction and that the discharged air has little or no tendency to spread the outside diameter of the fan.

Before considering the results of experiments made on fans of this type, it will be well to consider exactly what is their function and exactly what results are sought from them by the ventilating engineer.

It has been previously mentioned that a propeller type of fan is an extremely inefficient appliance when working against a pressure, and it can only be usefully employed in moving air against very low pressures. That it must work against *some* pressure is evident, for the mere movement



Figures denote Velocity in feet per minute

Mechanical Engineers 1897.

FIG. 72.

of the air implies that a certain pressure is and must be set up by the working of the fan, but the lower the difference of pressure can be kept on the two sides of the fan the more efficiently it will work.

When working with free inlet and discharge the vacuum set up on the suction side is just sufficient to generate the movement of the suction air towards the fan. The air is accelerated before reaching the fan and while between the blades. Little or no pressure is needed to accelerate the air in front of the fan, since the air is already moving with almost its maximum velocity the moment when it leaves the fan. Thus a propeller fan will work much better sucking than it will driving.

In a fan of this type the parts of the blades near the axis are moving at a much lower velocity than near the periphery. The velocity communicated to the air is therefore greater near the periphery than in the centre close to the axis, indeed, the blades are there practically producing no useful result. Therefore, if any pressure is generated in front of the fan, that is to say, if the outlet from the front of the fan is not freely open to the air the pressure generated will cause a return flow of air from the delivery side into the fan at the centre. At the periphery the blades will thus be choked by air which has already passed once through the blades.

That this is a true explanation of this phenomenon is proved by the fact that when the centre of the fan is blocked by means of a disc of considerable diameter so that air from the delivery side cannot find its way back into the eye, the efficiency of the fan as a pressure producing machine is very greatly increased.

It might be thought at first sight that the diagram of velocities, Fig. 72, proves that as much air enters the centre or eye of the fan as near the periphery. In fact, however, although this is true, the velocity is probably communicated to this central core of air partly merely by induction, and perhaps partly by the tendency to create a partial vacuum by the whirling motion of the air on the delivery side.

An important observation is that to obtain the maximum efficiency from a fan of this type, the outside of the blades should not be surrounded by any form of tube, but the fan should be placed several inches on the suction side of the opening through which the air is delivered, otherwise the tube round the blades merely serves as an obstruction to the air which would otherwise enter the fan round the periphery.

In considering the results obtained from such a fan as this it must be remarked that the main object is to obtain the largest delivery of air at the lowest cost in horse-power.

Mechanical Efficiency.

In order to investigate this matter it is necessary in the first place to consider what is the smallest possible amount of power mechanically necessary to move the air, and this depends on the velocity with which it is delivered.

A thousand cubic feet of air at 60 deg., weighing, say, 76.1 pounds, moving with a velocity v feet per second, necessarily involves the expenditure of $\frac{76.1\ v^2}{64.4}$ foot-pounds of work.

It is not conceivable that any lower expenditure of power than this should produce the result. It is clearly, therefore, advisable that the velocity should be as low as possible. The mechanical efficiency of the fan is the ratio of this power to that actually required to drive the fan when moving the quantity of air. From the diagram it will be seen that the actual local velocity of the stream of air varies considerably in different parts of the current. The value of v must be taken as the mean $\frac{Q}{a}$, where a is the area.

In practice such a fan is generally driven by an electric motor, steam engine, or other prime mover. The commercial efficiency is the ratio of the power in the air to the total power necessary to drive the motor, including all the losses in the motor itself.

It is very important to observe that what the engineer requires is to obtain the greatest delivery of air at the lowest cost in power. What really determines the power absorbed by the air is the velocity at which it is delivered. The velocity is a waste product as far as the engineer is concerned. He wants volume of air—not air at a velocity. A fan having a high mechanical efficiency would be an extravagant one if it discharged the air at a high velocity. A larger fan even having a much lower efficiency would be a much better proposition if it delivered the same volume of air at a lower cost in power owing to the lower velocity.

Volumetric Efficiency.

It is customary to calculate the volumetric efficiency on the same basis as has been explained for a centrifugal fan, *i.e.*, as the ratio between the amount of air actually delivered and the amount that would be delivered through a square flue whose side was equal to the radius of the fan if the mean velocity of the air had been equal to the tip speed.

In the opinion of the present Author a more rational interpretation of the expression, volumetric efficiency, would be the ratio of the quantity of air actually delivered to that which would be delivered if each revolution of the fan forced through the whole area of the fan a cylinder of air whose length was equal to the pitch of the periphery of the blades, and whose cross sectional area was equal to the area of the fan.

Pressure Efficiency.

What is generally taken to be the "pressure efficiency" of a propeller fan is on a similar basis to that already described for the case of centrifugal fans.

$$\text{Pressure efficiency} = \frac{v^2}{2\,u^2}$$

where u is the velocity of the tips of the blades. Unlike the same expression as applied to a centrifugal fan this function appears to be based on no scientific reasoning when applied to a propeller type of fan, but as a mere rule of thumb.

So far as the heating engineer is concerned, as has been stated above, what he requires is the maximum volume of air delivered at a minimum of power consumption and at a minimum first cost of the fan itself. The pressure generated is an extraneous consideration which reduces largely the efficiency of the fan, whose magnitude cannot be satisfactorily measured.

The ordinary rules governing the operation of fans of this kind are as follows :—

1. For a constant diameter, angle of blades, and position of the fan, the air discharged varies as the speed.
2. The twisting moment on the fan shaft varies as the square of the fan speed.
3. The horse-power varies as the cube of the speed.

These relations may also be written :—

For a given size of fan the revolutions vary as the air discharged.

The torque, or twisting moment, varies as the air discharged squared.

The horse-power varies as the air discharged cubed.

If Q is the quantity of air discharged in cubic feet per second, the horse-power *in the air*

$$\frac{v^2\,Q\,B}{T} \times 0.00003756$$

where B is the height of the barometer in inches of mercury and T the absolute temperature.

The mechanical efficiency of the fan, when the barometer reads 30″ of mercury and the air temperature is 60° F., for ordinary calculations may be taken generally to be about 30%.

The horse-power to drive the fan then becomes

$$v^2\,Q \times \frac{0.00000215}{0.3}$$

or since

$$v \times \frac{\pi}{4} D^2 = Q \text{ or } v = \frac{4\,Q}{\pi\,D^2} = 1.27\,\frac{Q}{D^2}$$

horse-power

$$\frac{Q^3}{D^4} \times 0.0000115$$

It is important to observe from these relations how greatly the horse-power increases when a smaller fan is set to do the same duty as a larger one.

The above paper quotes the power necessary to drive fans of five different diameters, each propelling 6,000 cubic feet per minute, and the table is as follows :—

TABLE SHOWING INCREASE OF POWER REQUIRED WITH SMALLER FAN FOR DELIVERING SAME QUANTITY OF AIR (*W. G. Walker*).

Diameter. Ft.	Horse-power.	Velocity of Air. Ft. per min.
2	0.72	1,910
2½	0.29	1,224
3	0.14	848
3½	0.071	623
4	0.045	477

Every special type of blade produces its own special result, and the only shapes for which well defined results can be given are fans consisting of a central boss and blades which are either plain or concave convex,

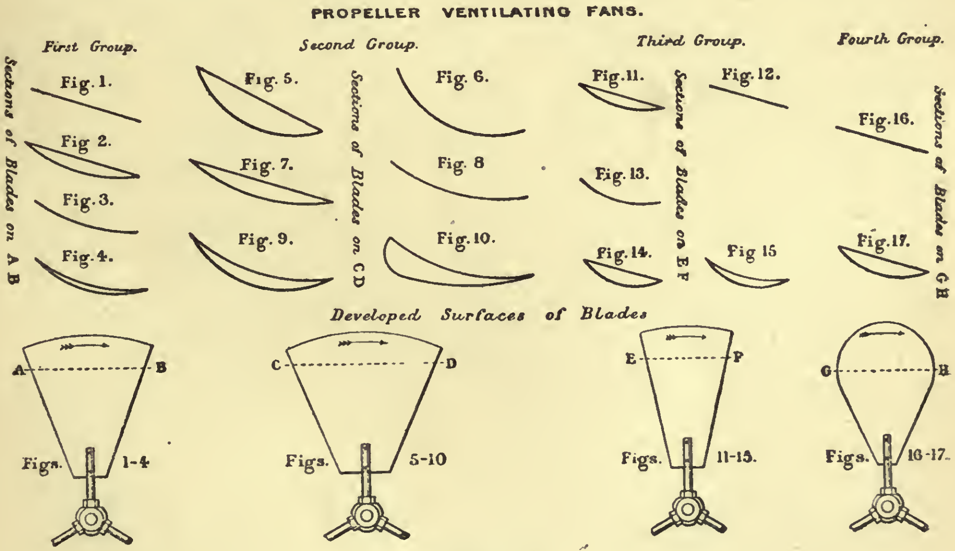


FIG. 73.
Scale: Blades 1 to 10 — 9″ wide.

which are straight in a radial direction, formed, that is, from flat blades bent into a cylinder. For fans of this type a very valuable series of experiments were reported by Mr. W. G. Walker in the paper alluded to above.

The results obtained in this series of experiments give a fairly complete view of the results which may be obtained from fans of this type, but the paper itself is somewhat marred in the opinion of the present writer by too close an adherence to the theory of the centrifugal fan in the method of calculating the results.

According to the present view the propeller fan and the centrifugal fan are essentially different in principle and should not be treated according to the same theory.

Effect of Shape of Blades.

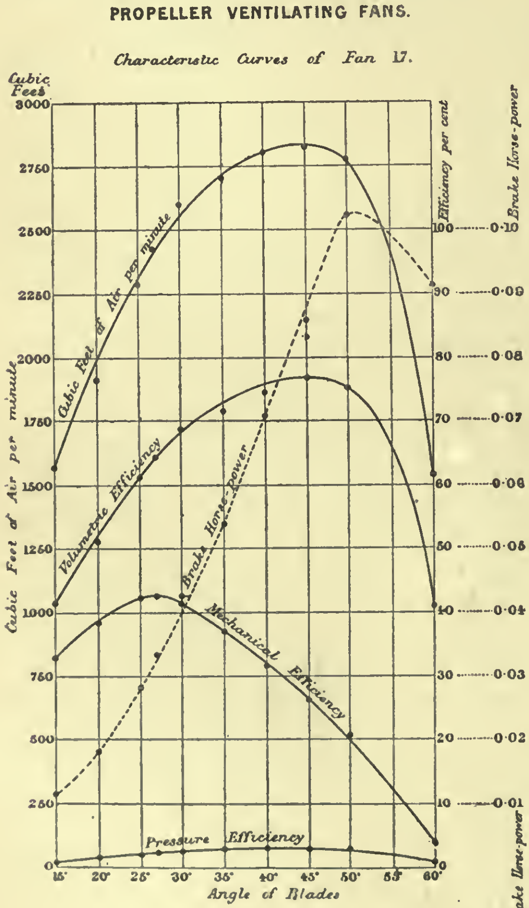


FIG. 74.

Careful experiments were made on this matter; the diagram, Fig. 73, shows sizes and shapes of all blades experimented on. These blades were also set at angles varying from 15° to 60°, but most of the experiments were made between 17° and 40°.

The table in the Appendix shows the effect of the different shape of blades working under more or less similar conditions.

Effect of Cross Section of Fan Blades.

The experiments alluded to above were made on blades of 17 different cross sections, having different widths and shapes. The general result was that blades of a curved shape were more efficient than the blades of a flat shape and that the tapered thick blades were more efficient than the single plate blades. The volumetric efficiency, as described above, and the mechanical efficiency are given in the table in the Appendix.

Effect of the Rotation of the Fan within the Tube.

A considerable reduction in the air delivery was noticed when the fan was allowed to rotate within a tube instead of having its edges clear. The figures in the table illustrate this point, from which it will be seen that by allowing the fan to rotate in an open space rather than within the tube increases the air delivery by some 50%.

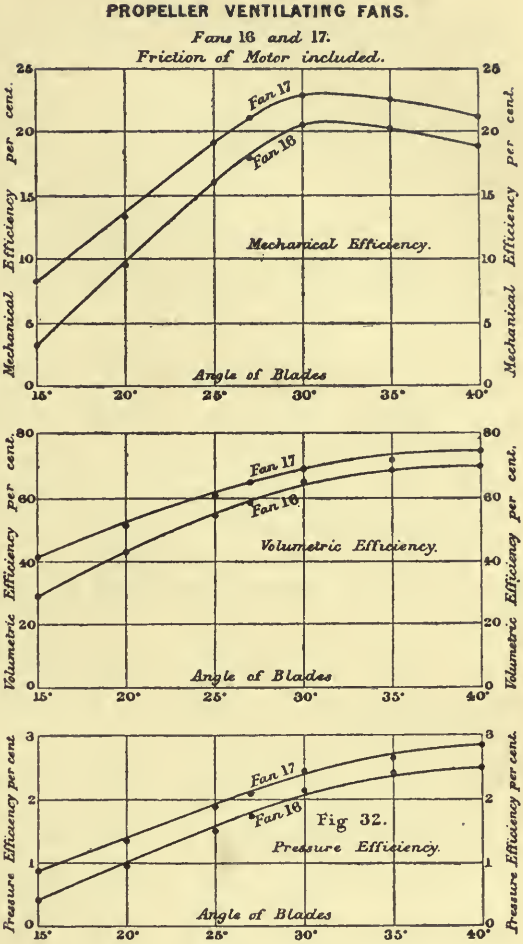


FIG. 75.

Effect of Alteration of Angle.

The most effective angle for maximum delivery of air is 45°. The diagrams Figs. 74 and 75 show how this increases as the angle increases. It is to be noted, however, that the maximum volumetric efficiency is not exactly the same as the maximum mechanical efficiency. Neither of these functions gives exactly what the heating engineer wishes to know, which is the conditions under which the fan will deliver the maximum quantity of air per unit of power.

In determining this, however, regard must be had to the characteristics of the motor which is being used. This is a matter which cannot be adequately dealt with in the present volume.

Effect of Number of Blades.

The results of Mr. Walker's experiments show that a six-blade fan is less efficient and discharges less air than a three-blade fan, and has less efficiency, and that little or no advantage can be gained by increasing the number of blades beyond the number of three.

Effect of Helical Blades.

These are blades of uniform pitch at all diameters. It was found that there is no noticeable advantage in adopting a helical as against a flat form of blade, and that the efficiency of a helical fan was increased by putting rounded backs on the blades in just the same way as with ordinary flat blades.

Approximate Calculation of Horse-Power and Size of Fan.

The horse-power necessary to drive the fan for a given discharge of air, assuming the height of the barometer is 30″ of mercury and the air temperature 60° F., is for ordinary purposes as follows :—

Assume a mechanical efficiency of 0.3 = horse-power in air discharged divided by horse-power to drive the fan. Then horse-power to drive fan =

$$V^2 Q \times \frac{0.00000215}{0.3}$$

or substituting the value of

$$V = \frac{Q}{A} = \frac{4 Q}{D^2} = \text{horse-power } \frac{Q^3}{D^4} \times .0000115$$

Assume a volumetric efficiency of 90%

$$0.9 = \frac{Q}{V R^2}$$

These values may be approximately obtained with fans working under ordinary practical conditions.

Pitot Tubes.

It has been already explained that when air moves along a flue every pound of it has a certain energy, due to its velocity only, and this energy per pound has been called the "velocity head."

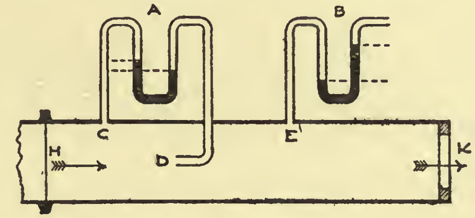
The velocity head, or kinetic energy, of one pound of the air is numerically equal to the height to which a body would attain if projected with that velocity vertically upwards, before it was brought to rest by the action of gravity.

The pressure energy on the other hand is the name given to the potential energy, which the body would have if it were raised up to the top of an imaginary column of the same fluid necessary to produce the pressure which it actually has.

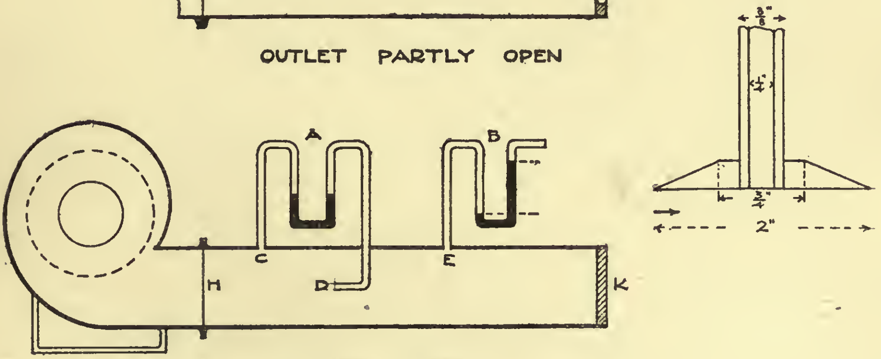
The simplest methods of measuring the velocity head and the pressure head in a stream of air, consists of what is known as a Pitot tube.

If a stream of air, having a velocity V, is directed normally against a plane surface, the pressure theoretically generated by the reaction is that which would be produced by a column of air $\frac{V^2}{2\,g}$ feet high.

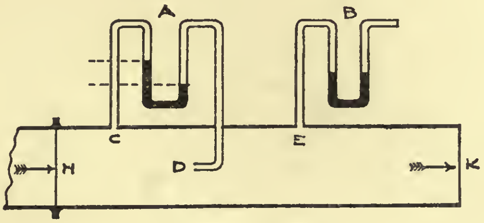
If, therefore, a tube is arranged, as in Fig. 76, having its opening facing the current of air, the pressure as read on a water gauge connected with the tube will represent the sum of the pressure of the air within the tube and the velocity head. If then the other limb of the water gauge is connected into the same flue, its outlet being perpendicular to the current of air, the pressure communicated to the corresponding limb of the tube



OUTLET PARTLY OPEN



OUTLET CLOSED



OUTLET OPEN

FIG. 76.

will be that due to the pressure of air only. The difference between the readings of the two limbs will therefore represent the velocity head.

The accuracy of a Pitot tube as an instrument for measuring these quantities was investigated by the Prussian Mining Commission in 1884. The result of the investigation showed that to obtain correctly the quantity of air by means of Pitot tubes, the following formula gives the velocity in metres per second at a temperature of zero centigrade.

$$V = 4.265 \sqrt{\frac{\text{pressure in MM. head of water}}{\text{density of air}}}$$

or transformed into British units

$$v \text{ in feet per sec.} = 17.6 \sqrt{\frac{\text{pressure in ins. water column}}{\text{density in lbs. per cubic foot}}}$$

It is also to be noted that if the absolute pressure is measured by means of a plain tube, the plane of whose outlet is parallel to the velocity of the air, it is apt to read incorrectly owing to the generation of dynamic pressure. This tendency can be overcome by the application of a flat plate of considerably larger diameter than the outlet of the tube, whose face is flush with that outlet in the manner shown on the right hand side of Fig. 76.

CHAPTER XII.

ON THE TRANSMISSION OF HEAT.

Conduction.

THE conduction of heat is of all phenomena that with which the heating engineer is most concerned. It is therefore very desirable that an accurate conception of the laws governing it should be obtained.

It consists, of course, of the communication by simple contact of that form of energy which we call heat, in an invisible manner from one body to another at a lower temperature. In its simplest form it consists of the communication of heat from a part of one body where the temperature is higher to other parts of the same body where the temperature is lower. It is to be distinguished from the process of convection, whereby heat is conveyed from one place to another by the transference of hot particles to another locality. Ultimately the two phenomena are identical as regards the particles, but not as regards the whole mass of substance, which consists of a collection of particles or separate bodies.

Mechanism of Conduction.

The heat existing in any body probably consists of a vibratory motion of the particles, or molecules, of which all bodies are supposed to consist. The particles of a body at a higher temperature are supposed to have vibrations greater in amplitude and more frequent in period than particles of the same body at a lower temperature.

Conceive a flat vertical plate of any substance which is heated at one side, say the right side. The particles of the substance on that side, by the process of heating, become more violently agitated than the rest of the particles further away from the source of heat. This agitation gradually communicates itself to the particles in a layer further to the left, which in turn increase in energy of vibration. The particles in this layer again communicate the agitation still further to the left, and so on throughout the whole body.

The propagation of the disturbance is naturally a slow process. The rapidity of the process depends partly on the readiness with which the particles take up the vibrations, and partly on the readiness with which they communicate the movement which they have absorbed to the adjacent particles. Generally we conceal our ignorance of the real nature of the phenomenon by saying that the rate of conductivity "depends on the nature of the molecules"—a non-committal expression.

The transmission of heat by conduction can, however, be investigated experimentally, and from the observations, rules can be obtained whereby the amount of heat passing in certain cases can be calculated with more or less accuracy.

What we really wish to get at, for practical purposes, is the amount of heat in thermal units which will be transmitted through a given kind of substance of a given thickness *from one medium to another*. This is not such a simple proposition as it looks. In practice the body or medium which contains the heat which is to be transmitted to another different kind of body is of a different substance from the plate (generally of metal) through which the heat is to be passed. The necessity for the communication of the heat from one fluid substance to another solid one, and from that solid again to a second fluid, gives rise to the question, "What is the effect of the separation of different substances on the flow of heat?"

The ultimate object being the transference of heat from one fluid to another, the conductivity of the body through which the heat passes is only of importance as affecting the ultimate result, and the first object of the investigation is to find out to what degree a difference in conductivity of the solid plate does actually affect the result.

Pure Conduction of Heat.

In the first place, in order to eliminate all questions as to the communication of heat from one substance (*e.g.*, water or air) to another substance (*e.g.*, iron), or vice versa, consider the case of a flat plate of considerable thickness, the *substance* of which is somehow or other maintained at a temperature t_1 in one plane, and in another parallel plane, at a distance e feet away from the first plane, the lower temperature t_2 is somehow or other maintained, and a constant regular flow of heat is thereby established between the two planes (see Fig. 77).

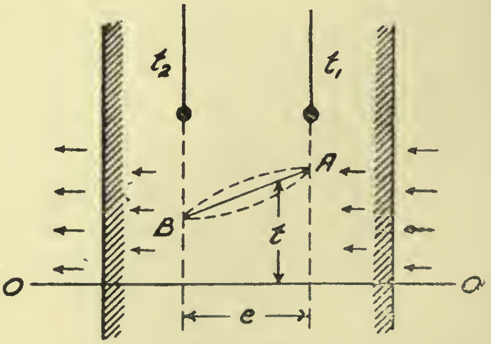


FIG. 77.

We have here the simplest case of pure conduction of heat through the substance of the material without having the transmission of heat from one substance to another to consider. The temperature may be conceived of as measured by means of two minute thermometers in two small holes drilled into the substance itself, one in each plane.

In these circumstances it has been proved experimentally, that the rate of transmission of heat is very nearly proportional to the difference of temperature, where the difference of temperature is not very great, that is to say, the quantity of heat H transmitted across unit thickness of the plate in a unit of time through a unit of area is

$$H = K\,(t_1 - t_2)$$

where K is a co-efficient to be determined by experiment, whose numerical value depends (1) on the quality of the substance of the plate, (2) on the units of length and time in which the other factors are expressed.

Where A is the area in square feet, and T the number of units of time in hours, during which the heat is conducted, the thickness being unity or one foot, the formula becomes

$$H = K\,(t_1 - t_2)\,A\,T.$$

It is also found that the temperature falls *gradually and almost evenly* from one of these planes to the other. If a hole were drilled half way between the planes, a thermometer placed in it would indicate a temperature *almost* exactly half way between t_1 and t_2, so that we could draw a "curve" of temperature $A\,B$ (in this case a straight line), whose height t at any point would show the temperature at that point in the manner shown in the figure.

The rate at which the temperature falls per unit of thickness of the plate is called the "temperature gradient." Mathematically it is expressed as follows:—

If $d\,\theta$ is the difference of temperature between two planes an infinitely small distance $d\,e$ apart, $d\,\theta\,/\,d\,e$ is the "temperature gradient." This value would be absolutely constant if the temperature fell *quite* uniformly from one plane to the other. The "temperature gradient" would then be numerically equal to the difference of temperature between two planes unit length apart. In any case the difference from uniformity is so small that it is taken to be equal to this difference for practical purposes.

K is numerically equal to the number of thermal units flowing in one hour (or other unit of time) through a unit area (square foot) of the plate, which has unit thickness (one foot), and which has 1° F. difference of temperature between its faces, or in other words, K is *the rate of flow of heat where the temperature gradient is unity.*

K is called the "conductivity" of the material.

It has been assumed in the above expression for K that the value of K is constant, whatever the temperature. Accurate investigations, however, have shown that K is not absolutely constant for different temperatures, and it does not even obey a constant or uniform law for different substances, so that the curve of temperature plotted on a base line representing thickness of material is not absolutely straight. It is, however, sufficiently near to a straight line for practical purposes.

Thus, for iron, the conductivity decreases as the temperature increases, whereas with copper the conductivity increases with the temperature. For instance, at a temperature of, say, 400° F. one square foot of iron plate will pass a smaller quantity of heat for a given difference of temperature than at a temperature of 60° F. On the other hand, a square foot of copper at the higher temperature will pass more heat per degree difference than it will at a lower temperature. The magnitude of the variation may be seen from the table below. The effect of this fact on the curve of temperature within the substance of the metal is that the true curve would not be an exact straight line like $A\ B$, but in the case of iron would be slightly convex with the convexity downwards, whereas with copper the convexity would be upwards, as shown in dotted lines on Fig. 77.

VARIATION OF CONDUCTIVITY WITH TEMPERATURE.

ABSOLUTE CONDUCTIVITY.

Temperature.	Iron.	Copper.
32°	38	203
212°	32.7	211
392°	29	220
572°	26.8	227

This fact, of course, would seriously complicate practical calculations if it were necessary to make them with scientific accuracy. But it will be shown in the course of the present investigation that the absolute conductivity of the material is, relatively speaking, of small practical importance, and therefore we may take for practical purposes the conductivity of each metal as a constant.

The Effect of the Thickness of a Plate on the Amount of Heat Conducted Through it.

Suppose *the substances* of the two surfaces of two plates A and B, Fig. 78, of the same material, and of which A is the thicker, to be each maintained at t_1 and t_2 respectively, the plates having thicknesses e_1 and e_2 respectively. By the law which has been previously explained, the temperature in each case will fall almost uniformly between the two surfaces.

When a uniform flow of heat has been established and all parts of either plate have attained a steady temperature (neither rising or falling) the same number of B.T.U. per hour must pass through any plane in that plate parallel to the boundary planes. If this were not so the heat would either accumulate in or leave some parts of the material, which is contrary to the assumption of a steady temperature.

Consider an imaginary plane cutting off from the thicker plate A a plate of the same thickness as the thin plate B. It is clear that the amount of heat passing per hour through this imaginary slice of A must be the same as is passing through the whole plate, for the rate of flow of heat is not altered by imagining a plane section passing through it. It is also seen from the diagram that the difference of temperature between the two faces of this imaginary section of A will be only e_2/e_1 of the whole difference in temperature $(t_1 - t_2)$.

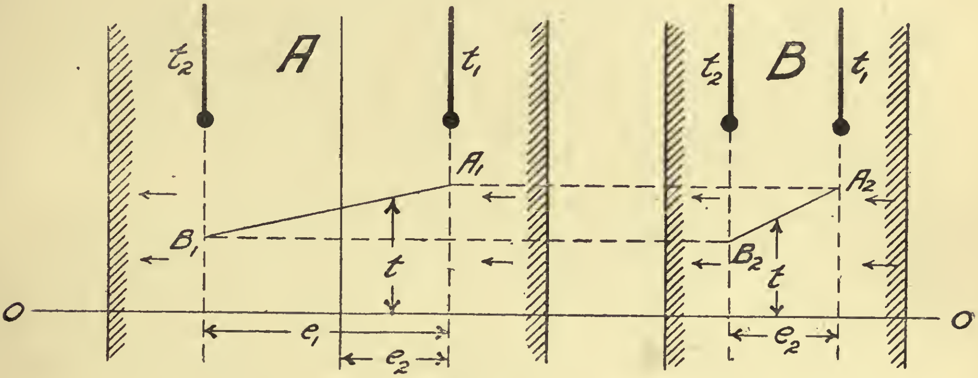


FIG. 78.

It was explained above that in any two plates of *equal* thickness the flow of heat is proportional to the difference of temperature.

Now compare the amount of heat H_1 flowing through the imaginary slice of A with H_2, the quantity, passing through plate B. The ratio of the quantities of heat flowing in the two cases must be proportional to the difference of temperature, that is

$$\frac{H_1}{H_2} = \frac{\frac{e_2}{e_1}(t_1 - t_2)}{(t_1 - t_2)} = \frac{e_2}{e_1}$$

or

$$\frac{H_1}{e_2} = \frac{H_2}{e_1}$$

This shows that the quantity of heat flowing through a plate when *the substance of the material* is maintained at a definite temperature in each case is inversely proportional to the thickness of the plate.

Note, however, the condition that the temperature is supposed *to be taken in the substance of the material itself*. t_1 and t_2 have nothing whatever to do with the temperature of the *media* in contact with the two sides of the plate which actually maintain the difference of temperature, but they are supposed to be actually *the temperatures in the substance of the material itself*.

The general equation, therefore, giving the total quantity H of heat conducted in T hours through a plate, of which the thickness is e feet, and area A square feet, of which one surface is maintained at t_1 and the other at t_2, and whose conductivity is K, is

$$H = \frac{K\,(t_1 - t_2)\,A\,T}{e}$$

The table in the Appendix gives the values of K or the actual conductivity of various materials.

EXAMPLE.—A plate of copper is 3/8″ thick, the actual temperature of *the surface of the copper* on one side is 112° F., and on the other side 90° F. Determine the quantity of heat which will pass through 3 square feet per hour.

Taking the figures from the table, we see that the quantity is

$$H = \frac{203 \times (112^\circ - 90^\circ) \times 3}{.0313} = \frac{203 \times 22 \times 3}{.0313} = 430{,}000 \text{ B.T.U.}$$

Conduction from Fluid Medium to a Solid Surface or vice versa.

Suppose the heat is communicated to the plate on one side by a body of hot fluid, whose mean temperature is θ, and that the temperature of the side of the plate in contact with this fluid is t_1, how much heat per unit of time will be communicated from the fluid to the surface of the material?

In practice this is a very complicated question, and definite figures cannot in all cases be given for it, which are of any use to the engineer. For a given difference of temperature it depends on the nature of the surface of the solid material, and on the nature and velocity of the fluid. If the surface is rough and the velocity relatively high the communication of heat will be relatively great. If it is smooth and the velocity low the communication of heat will be reduced. If the fluid is at rest it will be at a minimum.

Similar observations would hold in regard to the communication of heat from the surface of the solid material on the other side to a cooling medium at a lower temperature. The laws, too, vary according to the nature of the media on the two sides of the plate.

In some cases investigations can be made which are very useful to the engineer, enabling the results of individual experiments to be applied to a great variety of cases with some accuracy. In other cases, though precisely the same laws are involved, the impossibility of defining the actual conditions which obtain in practical work, and the great variations in the results which are brought about by variations in these conditions, render practically useless any definite laws which may be formulated, except as applied to very special cases, such as are extremely rare in practice. This matter can, in practice, only be dealt with by strictly practical experiments, which to some extent leave the exact theory out of account. This matter will be further discussed later in the present chapter.

The heat communicated to a fluid by any warm surface or to any cold surface by warm fluid really consists of two parts.

Consider first the heating of cold fluid by a hot surface. In this case the heat may pass from the hot body—

(1) By radiation.
(2) By conduction to the fluid in contact with the surface, the warmed fluid being subsequently distributed by convection currents.

In the present stage of the science of heating both these parts are regarded as identical, for both parts are assumed to find their way into or out of the room, although by different channels.

1.—Radiation.

The part radiated is not directly communicated to the fluid. It exists at first as a form of energy in the ether, and operates in raising directly the temperatures of the walls, furniture, and surrounding bodies without using the air as a carrier of heat.

The rays are propagated by the identical mechanism with which light rays are transmitted. This mechanism consists of waves of vibrational energy moving through the ether, at a velocity of about 186,000 miles per second. They are assumed to be generated by the vibration of the molecules of the body emitting heat, and are given off by all bodies alike, whatever their temperature. But a hotter body sends out in this form more energy than does a colder one.

Between any two bodies at the same temperature placed in juxtaposition there is a mutual interchange of these rays, each body receiving and absorbing from the other the same amount of energy as it emits, so that on the whole there is no gain or loss of heat by either body.

The passage of the rays does not raise the temperature of the air through which they pass. When they impinge on any absorbent surface or pass into any fluid which is opaque to them they are suppressed, their energy being converted into heat, which is absorbed by the receiving surfaces. Part of this heat may be given up to the air in contact with the absorbing surface.

If the surface is of a reflecting character the energy is partly radiated back, according to the same laws as those which govern the reflection of light rays.

When several bodies of different reflecting power, but at the same temperature, are placed within an enclosure at the same temperature, the exchange of rays does not raise or lower the temperature of any of them. Since bodies having a reflecting surface do not *absorb* so much of the radiation which impinges on them as do bodies with a dull black surface, it is plain that the former must *emit* proportionately fewer rays. If it were not so the bodies with a dull surface would rise in temperature and the reflecting bodies would fall. Just in so far as any surface is a reflecting one, to exactly the same extent it has a lower emissive power.

Diathermancy.

The rays are capable of passing through certain substances (of which air is one) which are more or less transparent to them, losing in the passage a certain proportion of their energy, varying with the nature of the substance. This portion is absorbed by the substance and converted into heat, and operates to raise the temperature of the partly transparent body.

The total proportion of the radiant energy which passes through thin plates of different materials varies according to the material. A thin plate of rock salt, for instance, transmits 92% of the incident radiation. This substance is probably the most transparent to heat rays, or as it is called, "diathermanous," of any solid substance known.

Plate-glass of about ⅜″ thick transmits 53% of the incident radiation when the temperature is high, such, for instance, as that of an open fire, but only a fractionally small quantity when the temperature is relatively low, such as that of a radiator.

Thus, that part of the incident radiant heat from a radiator which passes through glass is first absorbed by the glass and afterwards transmitted by conduction, being subsequently lost from the outer surface of the glass by convection.

A clear plate of rock salt, on the other hand, transmits the same proportion of radiation at all temperatures, namely, about 92%.

A layer of distilled water ⅜″ thick transmits about 11% of the heat from an argand lamp with a glass chimney.

Most diathermanous bodies exhibit this phenomenon of "selective absorption" as it is called. Certain of the rays or waves composing the beam of heat will pass easily through a certain diathermanous body, while other waves of different wave length will be absorbed by the same body. Such bodies are diathermanous to some rays, but opaque to others.

A very minute thickness of the substance will suffice to absorb all rays to which it is opaque. It is by the absorption of constituent rays that the total radiation transmitted is reduced, and therefore the thickness of the medium (within moderate limits) does not make much difference to the total absorption.

The laws expressing the results of investigation of the amount of radiation given off by bodies at different temperatures are of a highly complex character. The results obtained by different investigators also vary widely and are not at present of much practical use to the heating engineer.

It may be merely mentioned that the two general expressions which appear to be the most useful, giving the rate of cooling due to radiation only, are as follows :—

Stefan's Law.—M. Stefan concludes that the total *radiation* emitted by any body is proportional to the fourth power of the absolute temperature, or that

$$r = A \cdot (273 + \theta_1)^4 - A\,(273 + \theta_2)^4$$

where θ_1 is the temperature centigrade scale of the body and θ_2 of the enclosure in which it is placed, and A is a co-efficient depending on the nature of the body. The accuracy of this law is confirmed both by experiment and theoretically from the principles of thermo-dynamics and the electro-magnetic theory of light.

Peclet gives the law an entirely different form, as follows :—

$$r = 124.72 \times K \times 1.0077^{\theta}\ (1.0077^{t} - 1)$$

where r is the rate of cooling in calories per square metre per hour per degree;

K is a co-efficient depending on the nature of the surface;

θ is the external temperature of the surrounding bodies in deg. Cent.;

t is the excess of temperature of the radiating body over θ all in centigrade degrees.

This somewhat sketchy account of a highly involved and difficult matter may be supplemented by those interested by reading any advanced book on the theory of heat.

2.—Mechanism of Conduction to Air.

That part of the total emission of heat which is lost by conduction is conveyed to the walls and furniture in the following manner :—

The surface of the walls, or any of the contents of the room being cooler than the body of the air in the room, cools the layer of air in immediate contact with it, the layer thereby becoming more dense than the remainder of the air. It therefore sinks towards the floor and is replaced in contact with the solid body by another layer of warmer air. This fresh layer is also cooled in turn, thereby communicating part of its own heat to the surface, and so on continuously.

In exactly the same way the hot surface of a radiator makes the layer of air in immediate contact with it warmer and therefore less dense than the body of the air in the room. That layer therefore rises, and is replaced in contact with the radiator by a cooler layer, which in turn gets heated and rises out of contact with the hot surface.

The very presence, therefore, of hotter and colder bodies within a room causes a circulation of the air, thus distributing the heat.

The proportion of the total heat which is lost from the radiator as convection and conduction was experimentally investigated, among others, by Sir Isaac Newton, Dulong and Petit, and Peclet, to whom, in various degrees, our knowledge of the subject is mainly due. The general laws are as follows :—

The heat removed by convection is independent of the nature of the surface of the body and of the surrounding absolute temperature. It depends on the difference of temperature between the hot body and the air and on the velocity of the air in contact with the body, probably being proportional to the square root of the velocity.

Peclet gives the value for self-generated velocities in ordinary still air as follows :—

$$A = 0.552\,K^1\,t^{1.233}$$

t being the difference of temperature in degrees centigrade.

The value of K depends on the shape of the body, and is as follows in metric units :—

For vertical cylinder, radius r, height h, in metric units

$$\left(0.726 + \frac{0.0345}{\sqrt{r}}\right) \times (2.43 + 0.8758\sqrt{h})$$

For horizontal cylinder of radius r

$$2.058 + \frac{0.0382}{r}$$

For a vertical plane of height h

$$1.764 + \frac{0.636}{\sqrt{h}}$$

The following curve shows the ratio between the heat given off by radiation and convection, according to Peclet's experiments, for an ordinary 2″ c.i. vertical cylinder at different temperature.

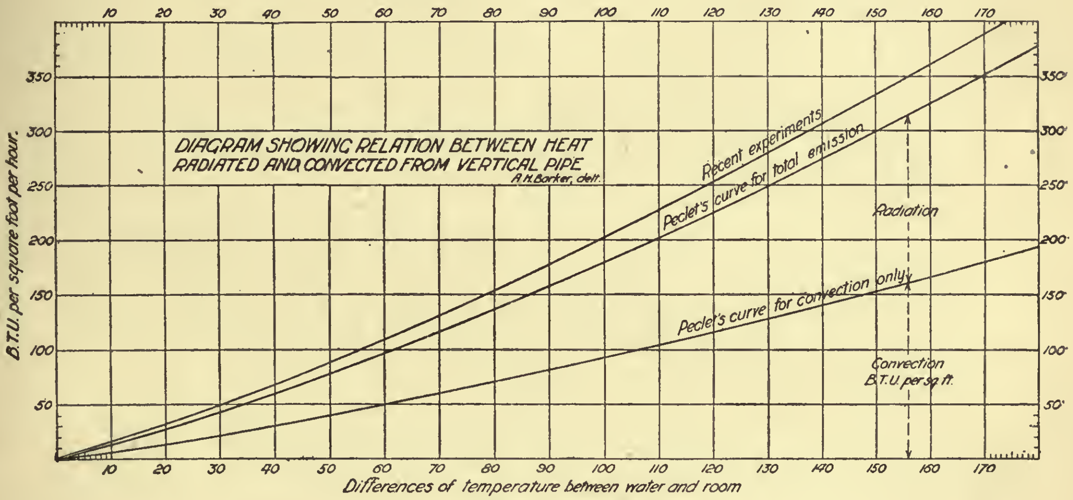


FIG. 79.

Applying these results to the case of a radiator it will be seen that the portion of the total heat emitted by radiation only is liable to great variations, according to the way in which the heating surface is massed together.

A large part of the surface of an ordinary radiator is exposed to radiation from other parts of the same radiator, and every ray which impinges on another part of the surface is absolutely lost or suppressed as radiant heat.

It is impossible to evolve any rules showing the amount of radiation so suppressed, but it is certain that with an ordinary radiator at least half of the entire amount of heat radiated is suppressed within the radiator itself. The proportion which escapes is approximately the ratio to the total surface of the radiator of twice the area of the front elevation + twice the area of the side elevation + twice the area of the radiator in plan.

For an ordinary two column radiator, having say 10 sections, 32″ high, this ratio would be about 58%.

The whole subject is surrounded with great difficulties. The investigation is probably one of the most difficult in the whole of experimental science.

In the present elementary stage of the science of heating it is still too obscure to be of first rate importance to the engineer.

The Communication of Heat through a Solid Substance from One Body of Fluid to Another.

When heat is transmitted from one body of fluid through a plate of solid substance, and to another body of fluid on the other side of the plate, the transmission of heat occurs in three stages :—

(1) The heat is communicated to the solid plate from the hot fluid.

(2) The heat is conducted from the warmer side of the solid plate to the cooler side.

(3) From the cooler side of the solid plate to the still cooler fluid on the other side of it.

Each of these three stages offers a resistance to the flow of heat and a separate co-efficient of conductivity applies to each. This point is further dealt with in the next chapter.

The amount of heat transferred across each of these resistances is found to be approximately proportional to the corresponding difference of temperature. Thus, the amount of heat transferred from a body of warm fluid to a cooler surface is proportional to the difference of temperature between the fluid and the surface.

It has been already shown that the quantity of heat transferred through a solid plate is proportional to the difference in temperature between the two surfaces, also the quantity of heat transferred from the cooler surface of the solid to the cooler fluid is proportional to the difference in temperature between them.

It will therefore be seen that the total quantity of heat transferred from the warmer fluid to the cooler fluid is proportional to the total difference of temperature between the two fluids.

The fundamental fact, on which depends the entire theory of the conduction of heat from one body of fluid to another, is that when the transmission of heat has been continued for a sufficient time under constant conditions through any system of solids and fluids, a constant flow of heat is ultimately established and the temperatures in all parts of the system assume values which are constant for those particular conditions, and further, that after that condition of thermal flow has been established the same quantity of heat passes in the same time across any section normal to the flow of heat.

If this were not so it is obvious that there must be either an accumulation or a loss of heat in or from some part of the system. For conceive two lenticular surfaces enclosing any heterogeneous part of a system of different bodies, and assume that a flow of heat is proceeding across the space enclosed by the surfaces. If the same nett quantity of heat does not pass surface 1 in any time as that which passes through surface 2 in the same time, it is clear that since heat cannot be destroyed or created during a process of simple conduction, if the flow of heat through the first surface were greater than that through the second one during the same time, there must be during the interval an accumulation of a certain amount of heat in the material between the two surfaces, which would cause a rise of temperature in the material.

This is contrary to the assumption of a uniform state of temperature.

A similar argument would hold good if the amount of heat flowing through the first surface were conceived to be less than that flowing through the second one.

In the case of heat passing from one hot fluid to a cold one through a solid, the evaluation of the temperature and the position of corresponding normal surfaces would of course become difficult, in the case of irregular surfaces, other than planes or cylinders, as would be also the determination of the stream lines of the heat. But even though the exact position of these lines, planes, and points cannot be determined, the truth of the theory is manifest that however irregular the surfaces may be, assuming that the heat has been passing for a sufficient time, the same amount of heat must pass from the hot fluid to the inner bounding surfaces of the solid as passes in the same time through the solid body, and from its outer bounding surface to the surrounding fluid.

In all cases in which the quantity of heat passing through a solid body from one body of fluid to another is in question, the only quantity the value of which can be treated as definitely determined, is the relation between the differences in the *body temperature*, the absolute conductivity and the thickness of the solid substance. This does not depend on anything except the nature of the material itself and conditions as to temperature which can be determined with some accuracy.

If, in any given case, we can determine these conditions with accuracy, we know definitely the amount of heat which must pass through the solid body, and therefore also the amount of heat passing into the solid body on the one side and out of it on the other side.

The fundamental difficulty is to determine figures applicable to a large variety of cases which will give with practical accuracy the quantity of heat flowing from the heating fluid into the solid on one surface, or from the solid to the *heated* fluid at the other. These quantities might conceivably be determined with some accuracy for one particular case, but the variations which are found to exist between different cases in different conditions are so wide that a result which might be accurate for one particular case would be widely in error in other cases to which the result might be applied.

Thus, the amount of heat flowing into a plate, say, of cast iron, from a body of hot water in contact with it on one side, depends not only on the difference in temperature between the layer of water in actual contact with the iron and that of the body of the iron itself. This alone is impossible to determine with accuracy on account of the fluctuations of temperature of the water. It depends also on the condition of the heat-receiving surface, and above all, *on the velocity of the water in contact with the plate*. Relatively, small differences in the velocity and temperature of the water can easily make a difference of 100% or more in transmission for the same difference of temperature.

Now in the great majority of practical cases it is next to impossible to determine accurately not only what is the condition of the heat-receiving surface, but also what is the velocity of the medium.

In cases where there are no special means provided for ensuring that the velocity of the water shall have any particular value, it is clear that when any reduction in temperature of the layer of water in close contact with the heat-receiving surface has been produced, owing to its heat having been transferred to the iron, the cool layer must hinder the flow of heat from the hotter water to the surface of the iron. The heat must be either conducted through the cool layer or the layer itself must be got out of the way by natural convection, brought about by the slight difference of density in the water, due to difference of temperature.

The velocity with which this process of convection will take place cannot be calculated, as it will depend on the form of the heating surface and the shape of the rest of the body of which the heating surface forms a part. Even if it could be calculated for any given case, that result would not be generally applicable to the wide variety of cases with which the heating engineer has to deal.

In cases, however, where there is a forced circulation of water over the surface, the layers of water cooled by contact with the surface are continuously scoured away from the surface, enabling the heat in the hotter water to get at the surface more readily. This is the reason why the transmission from a liquid to a solid surface is so much greater when the relative velocity between the liquid and the surface is considerable.

In cases where the heat is communicated to the solid surface by means of steam the conditions are considerably more constant. Steam is a much more mobile conveyor of heat than is water. As soon as a layer of steam in contact with the surface has parted with its heat to that surface it collapses into water. This water on its first formation forms a thin and relatively non-conducting layer on the inside surface, acting so as to protect the surface to some extent from the free flow of heat. When this layer of water attains a certain thickness, owing to the continuance of the process of condensation, if the surface is vertical or inclined it runs away of itself and makes way for a fresh supply of heat to reach the surface.

If the surface is nearly horizontal and *below the steam* it does not clear itself readily. Although the conditions under which the flow of heat from steam to the surface are more definite than in the case of water, yet even they are not sufficiently precise to enable mathematical deductions to be made from them.

As in the case of water, the transmission of heat from the steam to the surface is also greater as the velocity of the steam increases, on account of the same scouring action whereby the layers of condensed water are continuously torn away from the solid surface, and the heat in the steam is thereby enabled to reach the solid surface more easily.

A further matter which makes the experimental investigation of this problem so difficult is that in many cases the actual *difference in temperature* between the heating fluid and the surface itself may be very small and liable to fluctuate one or two hundred per cent. from time to time without any appreciable change in the conditions. This is particularly the case where the fluid on the one side of the surface is hot water or steam and on the other side air.

The rate of conduction from steam or water to iron is much greater than the rate of conduction from iron to air, and therefore in this case the actual temperature of the iron is much nearer to the temperature of the heating than of the heated medium. Not only so, but the small thicknesses of metal which are used in all practical cases make the determination of the actual temperature difference between the fluid in close contact with the metal and the metal itself almost impossible.

As has been shown, even if the laws could be accurately determined they would be of little practical use for the heating engineer. He cannot regulate his conditions as they can be regulated in the laboratory, and he must take materials and circumstances as he finds them.

It is relatively easy in some cases to determine *roughly* by experiment the *total* flow of heat from, let us say, steam through a vertical metal surface to air on the outside of the surface. Knowing the conductivity and thickness of the metal it is therefore easy to determine with some accuracy what must in consequence be the difference of temperature between the inner and outer surface of the metal. This is the utmost that can generally be done in exact scientific investigations of practical cases. The remainder of the information required by the engineer can only be obtained by more or less rough overall experiment, under conditions which are of necessity far removed from the scientific ideal.

Influence of the Conductivity of the Material and its Thickness.

It is easy, however, to show from this fact what is the relative importance of the nature of the material in such a case as a radiator.

Suppose, for instance, we are comparing two similar radiators, one made of brass and the other of cast iron, both having a thickness of ⅛", and that the radiators are working at a temperature at which they give off about 160 B.T.U. per hour per square foot. This corresponds roughly to an interior temperature of the water of about 175° F. The difference of temperature between the two surfaces of the iron radiator may be obtained as follows :—

From the formula

$$H = \frac{K\,(t_1 - t_2)}{e}$$

$$160 \text{ B.T.U.} = \frac{38}{.0104}\,(t_1 - t_2)$$

whence

$$t_1 - t_2 = \frac{160 \times .0104}{38} = .044^\circ \text{ F.}$$

For a copper radiator the difference would be

$$\frac{160 \times .0104}{203} = .0082^\circ \text{ F.}$$

For a brass radiator the difference would be

$$\frac{160 \times .0104}{54} = .031^\circ \text{ F.}$$

It will be seen from this that a difference in the metal of which a radiator is made only produces a difference in outside skin temperature of the metal of a small fraction of a degree. It is on the outside skin temperature of the metal that the communication of heat to the air mainly depends, apart, of course, from the velocity of the air in contact with the surface, which latter is the most important of all considerations.

A similar investigation may be made in the case of a steam pipe heating water. In this case let us compare the effect of the difference in conductivity of a wrought iron pipe, ⅛" thick, and a copper pipe of the same thickness, and of *an identical kind of surface*, other conditions also being identical. Suppose the temperature of the steam in the pipe to be 240° F., and the temperature of the water 160° F.—a difference of 80° F. Here the total rate of transmission of heat per square foot would be about

$300 \times 80^\circ = 24{,}000$ B.T.U. per square foot per hour.

This would produce a difference in the temperature of the metal itself. In the case of iron

$$\frac{24{,}000 \times .0104}{38} = 6.6^\circ \text{ F.}$$

and of copper, assuming the same heat transmission,

$$\frac{24{,}000 \times .0104}{203} = 1.23^\circ \text{ F.}$$

which is, at most, a relatively small difference.

The thickness of metal, too, only produces a comparatively small alteration in the difference of temperature. For the same rate of transmission this is nearly proportional to the thickness, the main factor, of course, being the temperatures and relative velocities of the water inside and the water outside the pipes, and the condition of the surface.

CHAPTER XIII.

THE TRANSMISSION OF HEAT *(Continued)*.

Practical Application.

IT now remains to apply the principles enunciated in the previous chapter to the transmission of heat in practical cases, as far as it can be usefully done.

Theoretical investigations are of the greatest value when applied to the case of the transmission of heat through outside walls of buildings from relatively warm inner air to relatively cold outer air. In this case the thickness of the solid wall is substantial, and the differences in temperature and the conditions generally are such that experimental investigations can be much more easily made than in the case of metal plates heated by water or steam. Also the quantity of heat passing from the inner air to the wall, and through the wall to the outside air is not only much smaller per unit area, but also more constant than where the heating fluid is water or steam and the material a thin plate of a relatively good conductor. The experiments, therefore, are much more manageable, and although the quality and therefore the conductivity of the solid wall is subject to much greater variation than is the case with a metal plate, yet the results that have been obtained by theoretical investigations in the case of a wall having air on two sides of it are much more accurate in practice than is the case with experiments on the transmission of heat from water to metal.

The following investigation, though it is theoretically as applicable to the one case as the other, is, in practice, chiefly of use as applied to the conduction of heat through walls of buildings from air inside a building to air outside, and to some extent also from metal to air, and from air to air, through metal flues and the like.

Wall e Feet Thick Having Air on Both Sides of it.

In this case there are four temperatures to be considered—the temperature (1) of the warm air on one side of the wall, (2) of the warm surface of the wall, (3) on the cold surface of the wall, and (4) of the cold outside air.

These we shall call t_1, T_1, T_0 and t_0.

The thickness of the wall will be called e.

Heat will not be transmitted from warm air at a temperature t_i to a surface at T_i, unless t_i is greater than T_i. It is equally clear that heat will not pass from one surface of the wall to the other, unless T_i is greater than T_o, similarly that it will not pass from the outer surface of the wall unless T_o is greater than t_o.

When a constant flow of heat has been established, the amount of heat communicated from the warm air at t_i to the wall at T_i will be exactly equal to that which, owing to the difference of temperature $(T_i - T_o)$ passes through the substance of the wall, and this again is equal to the amount which passes from the outside of the wall at T_o to the outside air at t_o.

The only matter we have yet considered is the passage of heat through the brickwork of the wall with a constant difference of temperature between its faces $(T_i - T_o)$. As seen from the table No. II., this quantity will be

$$\frac{0.47}{e} (T_i - T_o) \text{ B.T.U. per square foot per hour.}$$

It has been proved experimentally that the amount of heat which passes from air to any surface in unit time is proportional to the difference of temperature between the air and the surface. Thus, the amount of heat passing from the warm air at t_i to the wall at T_i is

$$a\ (t_i - T_i) \text{ B.T.U. per square foot per hour,}$$

where a is a constant whose value depends on the character of the surface of the wall and on the velocity of the air over it.

The rougher the surface and the more absorbent the nature of the material with which it is covered, and the higher the velocity of the air passing over it, the greater will be the value of a. This value can only be determined by elaborate experiment.

Peclet, to whom our knowledge of this subject is mainly due, gives as the value of a the equation

$$a = l + s + (0.0075\ l + 0.0056\ s)\ (T_o - t_o)$$

for an outer wall and

$$a = l + s + 0.0075\ l\ (t_i - T_i)$$

for an inner wall, when quantities of heat are measured in calories per square metre per hour per degree Cent., and where

l is a factor whose value depends on the movement of the air.

s is a factor whose value depends on the condition of the surface of the wall and the material of which it is made.

In order to reduce these figures to B.T.U. per square foot per hour, per degree F., it is necessary to multiply the value of a by

$$\frac{3.968}{10.76 \times 1.8} = 0.205$$

or in other words, multiply Peclet's given values of l and s by 0.205, and divide the numerical co-efficients within the bracket by 1.8, to make up for the change in the value of the degree (see section on units at page 12), so that the equations converted into English units are

$$a_o = l + s + (0.00416\, l + 0.0031\, s)\,(T_o - t_o)$$

and

$$a_i = l + s + 0.00416\, l\,(t_i - T_i)$$

Rietschel, following Valerius and Grashof, gives for

VALUES OF l.

	C.G.S. Units.		Brit. Units.
Air at very low velocities, *e.g.*, air in contact with cold walls	4	corresponding to	0.82
Air at low velocities, *e.g.*, for air in contact with cold windows	5	,, ,,	1.02
Moving air, *e.g.*, outside air	6	,, ,,	1.23

VALUES OF s.

NON-METALS.	Brit. Units.	METALS.	Brit. Units.
Cotton Fabric	0.75	Rusty Iron	0.69
Oil Paint	0.76	Sheet Iron	0.57
Building Stone ...	0.74	New Cast Iron ...	0.65
Glass (dry)	0.60	Copper	0.033
Glass (wet)	1.08	Brass (Polished) ...	0.053
Wood	0.74	Silver	0.027
Chalk (Powdered) ...	0.68	Zinc	0.049
Water	1.09	Tin	0.045
Wool	0.76		
Paper	0.78		
Paper (Silvered) ...	0.086		
Paper (Gilt)	0.047		
Lamp Black	0.82		
Sand	0.75		
Sawdust	0.73		
Silk	0.76		

It will be easily seen that the last term in the equation for a must in any case be very small, for the values of l and s are both between 0 and 1.25 as extreme limits. The general values being about

$$l = 1 \qquad s = 0.75$$

Also

$$(t_i - T_i) \text{ or } (T_o - t_o)$$

cannot in the extreme case be greater than say 20°, and assuming approximately $l = s = 1$ (also extreme values), the value of the last term is roughly about

$$= 0.14$$

so that in any case the last term cannot be greater than a small fraction, which, in an approximation of this character, is often negligible compared to the value of $l + s$. The values of l and s are given in the table above.

We are seeking to obtain a co-efficient K for the *entire* transference of heat from the air inside to the air outside, such that $H = K(t_i - t_o)$ in B.T.U. per square foot of wall, that is, a co-efficient which when it is multiplied by the difference of temperature between the air inside and outside the building will give the number of B.T.U. per hour which will pass through every square foot of wall.

The total quantity of heat transmitted per square foot per hour

$$= K(t_i - t_o) = a_i(t_i - T_i) = \frac{k}{e}(T_i - T_o) = a_o(T_o - t_o)$$

which is an expression of the fact that the quantity of heat which passes into the wall must be equal to the quantity of heat which passes through and out of the wall on the cold side.

The following investigation shows how the value of K depends on that of a_i, a_o, k and e.

In examining this expression it will be well to investigate the effect of a variation in the value of the different factors which go to make up the value of K, for that is the main practical object which this investigation has. It will be seen that the greater a_1 and a_o are, the greater K will become.

To Find Value of *K* for Single Wall.

We have

$$a_1(t_i - T_i) = K(t_i - t_o)$$
$$a_o(T_o - t_o) = K(t_i - t_o)$$
$$\frac{k}{e}(T_i - T_o) = K(t_i - t_o)$$

whence

$$t_i - T_i = \frac{K}{a_1}(t_i - t_o)$$
$$T_o - t_o = \frac{K}{a_o}(t_i - t_o)$$
$$T_i - T_o = \frac{eK}{k}(t_i - t_o)$$

Adding these three equations together

$$t_i - t_o = K(t_i - t_o)\left(\frac{1}{a_1} + \frac{1}{a_o} + \frac{e}{k}\right)$$

whence transferring

$$K = \frac{1}{\frac{1}{a_1} + \frac{1}{a_0} + \frac{e}{k}} \quad \text{or} \quad \frac{1}{K} = \frac{1}{a_1} + \frac{1}{a_0} + \frac{e}{k}$$

From which equation, along with the above values of a_i and a_o and k, the conduction through any wall can be calculated.

Find Values for *K* for Compound Walls.

An exactly similar investigation shows that if a wall is composed of n touching layers

$$K = \frac{1}{\frac{1}{a_1} + \frac{1}{a_0} + \frac{e_1}{k_1} + \frac{e_2}{k_2} + \frac{e_n}{k_n}}$$

Whereas if a wall has an air space,

$$K = \frac{1}{\frac{1}{a_1} + \frac{1}{a_0} + \frac{1}{a_{i1}} + \frac{1}{a_{o1}} + \frac{e_1}{k_1} + \frac{e_2}{k_2}}$$

and so on.

For a glass or thin metal wall the thickness of the plate is so small that $\frac{e}{k}$ becomes practically negligible, hence we see that a metal sheet conducts practically the same amount of heat from air to air as a glass one having equally smooth surfaces. Also the value of s, for glass does not differ materially from that for sheet iron, so that the two results are practically the same, and experience bears it out, that a glass house loses about the same amount of heat as an iron one.

To Determine the Value of $t_i - T_i$.

This value can be obtained in any given case as follows :—

Having found the value of K from above equations, we have from equation three on previous page

$$t_i - T_1 = \frac{K}{a_1}(t_i - t_o)$$

If then a brick wall is, say, 18″ thick, we find the value of

$$K = 0.24$$

then

$$a_1 = l + s + 0.0075\, l\, (t_i - T_o)$$

The last term may be neglected in an approximation of this character.

$$a_1 = 0.82 + 0.75 = 1.57.$$

Assume $t_o = 32°$, and $t_i = 60°$, as usual in England.

$$t_i - T_i = \frac{0.24}{1.57} \times 28 = 4.25° \text{ F.}$$

The diagram shows the value of this difference calculated in this manner for different thicknesses of wall.

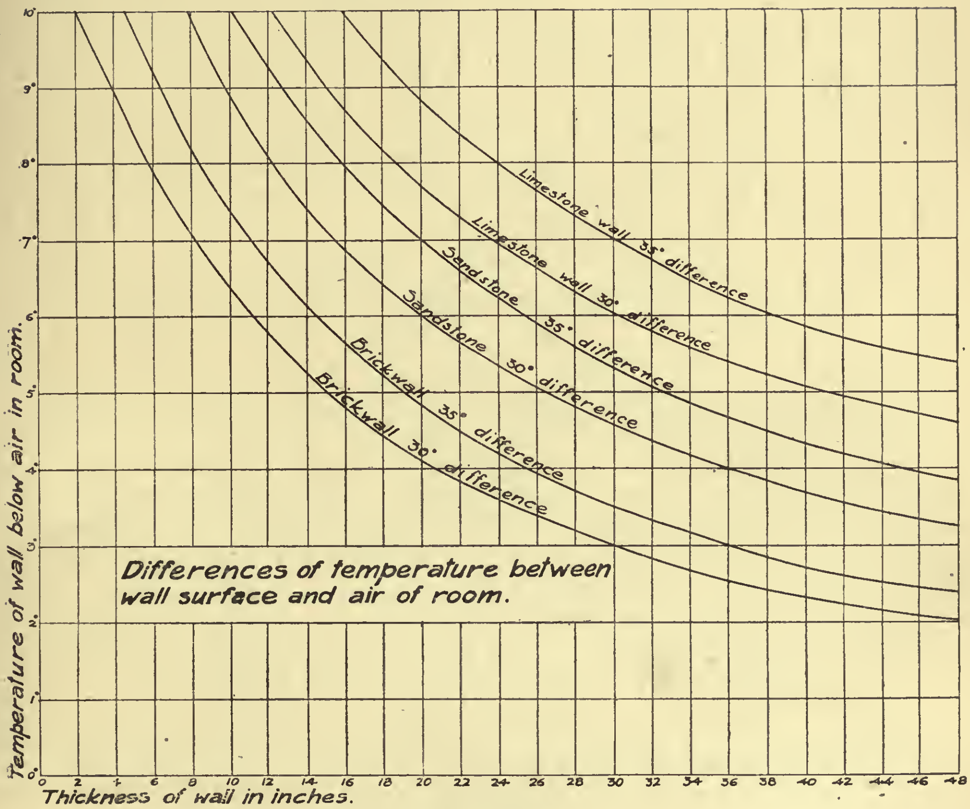


FIG. 79a.

Examples Showing Method of Calculating Value of Co-efficient of Transmission *K*.

1. Single sheet of glass of thickness 1/16″.

Formula

$$\frac{1}{K} = \frac{1}{a_1} + \frac{1}{a_o} + \frac{e}{k}$$

Value of a_i (see page 262)

$$= l + s + (0.00416\, l + 0031\, s)\,(t_i - T_i)$$

The inner surface of the glass is assumed to be wet with condensed vapour, the outside dry and exposed to wind, whence

$$s_i = 1.08 \qquad s_o = 0.60$$

The value of $t_1 - T_1$ in this case may be assumed to be half the total difference of temperature

$$= \frac{t_1 - t_0}{2}$$

or, say, 15° F.

The extreme possible error in making this assumption is small. Hence

$$\begin{aligned} a_1 &= 1.02 + 1.08 + (0.00416 \times 1.02 + .0031 \times 1.08)\ 15. \\ &= 1.02 + 1.08 + 0.114 = 2.214. \end{aligned}$$

Similarly

$$\begin{aligned} a_0 &= 1.23 + 0.60 + (0.00416 \times 1.23 + 0.0031 \times 0.60) \times 15^\circ \\ &= 1.23 + 0.60 + (.00512 + .00186) \times 15^\circ. \\ &= 1.23 + 0.60 + .1045 = 1.935. \end{aligned}$$

Hence

$$\frac{1}{K} = \frac{1}{2.214} + \frac{1}{1.935} + \frac{0.005}{0.55}$$
$$= 0.45 + 0.517 + 0.0091 = .976$$
$$K = \frac{1}{0.976} = 1.03$$

This is slightly smaller than the value commonly adopted (1.05), because that value was primarily calculated according to continental conditions, whence the extreme temperature is lower. This has an effect on the value of a_1 and a_0, which influences the resulting values of the equation.

1*a*. Compare the effect of plate glass windows, say ¼″ thick, with that of common glass 1/16″ thick.

Here the equation is the same except for the last term, which becomes

$$\frac{1}{K} = 0.45 + 0.517 + \frac{0.0208}{0.55} = 1.058$$
$$K = 0.95.$$

The difference is seen to be small.

2. The value of double windows of 1/16″ glass. The difference between inside and outside temperature is supposed 30°.

For an approximate calculation we may assume that the temperature of the air between the windows is the mean of the inner and outer temperature, or 15° above outside and below inside air. The effect would be approximately to halve the value of K for the whole transmission.

A more correct calculation would be as follows :—

It is necessary to find the value of a for the following cases :—

(1) Between the air in room and room surface of inside window.

(2) Between outer surface of inner window and the space between windows.

(3) Between the space between the windows and inner surface of outer window.

(4) Outer surface of outside window and outer air.

The inner window is here calculated as being dry and the outer one wet. The temperature of the inner window is calculated as being 7.5° below the inner temperature.

$$a_1 = 1.02 + 0.60 + (0.00416 \times 1.02 + 0.0031 \times 0.60)\,7.5^\circ.$$
$$= 1.02 + 0.60 + .046 = 1.67.$$
$$a_2 = 0.82 + 0.60 + (0.00416 \times 0.82 + 0.0031 \times 0.60)\,7.5^\circ.$$
$$= 0.82 + 0.60 + .395 = 1.46.$$
$$a_3 = 0.82 + 0.60 + (0.00416 \times 0.82 + 0.0031 \times 0.60)\,7.5^\circ = 1.46.$$
$$a_4 = 1.23 + 1.08 + (0.00416 \times 1.23 + 0.0031 \times 1.08)\,7.5^\circ.$$
$$= 1.23 + 1.08 + 0.063 = 2.373.$$

Here therefore

$$\frac{1}{K} = \frac{1}{1.67} + \frac{1}{1.46} + \frac{1}{1.46} + \frac{1}{2.373} + \frac{2 \times .005}{0.55}$$
$$= 0.6 + 0.685 + 0.685 + 0.42 + .0182 = 2.408$$
$$K = \frac{1}{2.408} = 0.416$$

which is slightly below the half of K as found by the rough approximation.

The student is advised to analyse for himself the effect on the value of K of assuming different conditions.

For example, compare in an extreme case the effect on the heat transmission through a single window, .005 foot thick, if (1) still dry air inside and outside, and (2) a high wind accompanied by rain with moist inner air.

Here the difference is chiefly in the values of a.

In the case of still dry air, we have

$$a_{01} = 0.82 + 0.60 + (0.00416 \times 0.82 + 0.0031 \times 0.60)\,15^\circ.$$
$$= 0.82 + 0.60\,(.00341 + .00186) \times 15^\circ = 1.499.$$

In the case of high wind and moist air

$$a_{02} = 1.23 + 1.08 + (0.00416 \times 1.23 + 0.0031 \times 1.08)\,15^\circ.$$
$$= 1.23 + 1.08\,(.00513 + .00335)\,15^\circ = 2.437.$$

In the case of still dry inner and outer air we have, therefore,

$$\frac{1}{K} = \frac{1}{1.499} + \frac{1}{1.499} + \frac{0.005}{.55}$$
$$= 0.67 + 0.67 + 0.009 = 1.347$$
$$K = 0.74$$

With the opposite conditions

$$\frac{1}{K} = \frac{1}{2.214} + \frac{1}{2.437} + \frac{0.005}{0.55} = 0.45 + 0.41 + 0.009 = 0.871$$
$$K = 1.15$$

or a difference of more than 50%.

The actual difference would, of course, be greatly increased by the rain running down the window outside, which would reduce the temperature of the window surface almost to that of the rain.

Assuming these conditions, it is clear that the total value of K would then be the total value of a_1 under the limiting conditions.

We should then have

$$\frac{1}{a_0} = 0 \text{ and } K = a_1$$

$$K = a_1 = 1.02 + 1.08 + (0.00416 \times 1.02 + 0.0031 \times 1.08)\,30^\circ$$
$$= 2.327$$

It will be seen, therefore, that the value of K for a single window varies from 0.75 to about 2.3, according to the conditions prevailing in the inner and outer air.

The diagram Fig. 80 shows the values of K calculated in the above manner for ordinary walls, etc., of buildings. These values are also given in Table XIV. in the Appendix.

On the Conduction of Heat from Steam or Water to Air Through a Metal Plate.

This is the aspect of conduction with which the heating engineer is perhaps more directly connected than any other, and it is at the same time the most difficult both to explain satisfactorily and to determine accurate co-efficients for. The difficulty mainly arises from the fact that the transmission depends very largely (1) on the way in which the surface is arranged, and the facility with which the heating and heated fluids can reach their respective sides of the surface, and (2) on the velocity of the fluids in contact with the surfaces, and especially on the velocity of the air.

Since heat is communicated much more readily from water or steam to metal than from the metal to the surrounding air, it follows that the actual body temperature of the metal is much closer to that of the heating medium than to that of the air.

It is therefore evident that the velocity of the heating medium cannot have as much effect on the total transmission as the velocity of the air, for there is a much smaller difference to play with. It is also clear from what has gone before that the difference in body temperature between the two sides of the metal must be very small.

The effect of increasing the velocity of air in contact with the outside of the metal is to reduce the outside skin temperature of the metal, and therefore to increase the temperature gradient within the metal itself, and also to some slight extent to reduce the inside skin temperature of the metal.

Now it is impossible to specify exactly or to measure in any given case the exact velocity of the layer of either fluid in contact with the metal

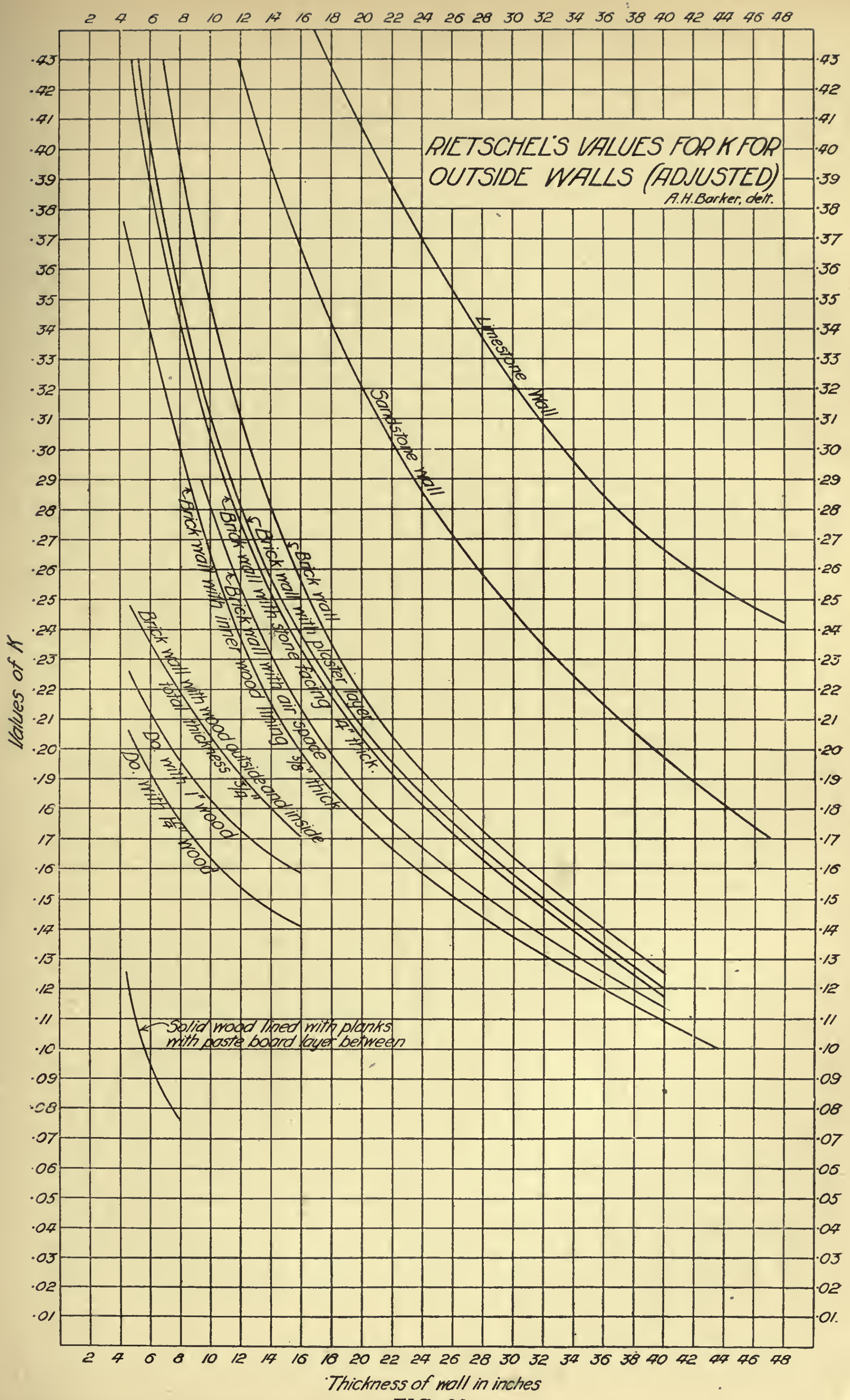

RIETSCHEL'S VALUES FOR K FOR OUTSIDE WALLS (ADJUSTED)
A.H.Barker, delt.
Limestone Wall
Sandstone Wall
Brick wall
Brick wall with plaster layer ¼" thick.
Brick wall with stone facing
Brick wall with air space ⅜" thick
Brick wall with inner wood lining
Brick wall with wood outside and inside total thickness ¾"
Do. with 1" wood
Do. with 1¼" wood
Solid wood lined with planks with paste board layer between
Values of K
Thickness of wall in inches
2 4 6 8 10 12 14 16 18 20 22 24 26 28 30 32 34 36 38 40 42 44 46 48
·43 ·42 ·41 ·40 ·39 ·38 ·37 36 35 34 ·33 ·32 ·31 ·30 29 28 ·27 ·26 ·25 ·24 ·23 ·22 ·21 ·20 ·19 16 17 ·16 ·15 ·14 ·13 ·12 ·11 ·10 ·09 ·08 ·07 ·06 ·05 ·04 ·03 ·02 ·01

FIG 80.

on either side, and as the rate of transmission depends very largely on the magnitude of this velocity, it is clear that any actual co-efficients with precisely definite conditions must be difficult or impossible to obtain.

In order to obtain a co-efficient under conditions which are at all comparable in scientific accuracy to those under which the conduction of heat within the solid body of a metal is determined, it is necessary to postulate that both fluid media shall be at rest, and at the same time, in order to evaluate these co-efficients in a satisfactory manner, it is necessary that the temperature shall remain constant, and these two conditions are mutually inconsistent.

It is not possible that the temperature of both media should be constant, and at the same time the condition of no movement be fulfilled, for if there is no movement of the media it is evident that the same layer of both the heating and the heated media must remain in contact with the surface. Any considerable transference of heat must therefore be attended by a diminution of the temperature of the heating fluid, and an increase of that of the heated fluid. This is contrary to the supposition of a constant temperature. The amount of heat which could be removed from the layer in contact with the metal by pure fluid conduction alone would be so extremely small as to make the experiment practically valueless.

In order, therefore, that the condition of constant temperature may be maintained, it is necessary to postulate a very slow motion of the media. This is unsatisfactory, because a very small difference in the velocity of either medium makes a very marked difference in the resulting value of *K*.

This scientific or ideal value of *K*, even if it could be accurately determined, would be of little use as applied to an actual case, for the conditions always surrounding any surface through which heat is being transmitted in practice vary so much that they cannot be either closely calculated or approximated in the laboratory.

The ideal scientific value of *K* then would represent the rate of emission of heat per square foot per degree difference for a very small surface, when the temperature of the heating medium and also of the heated medium are both constant, and when there is no motion of either medium.

The essential difference between this co-efficient and any practical one that could be determined experimentally is considerable. The nearest conditions which could be obtained to these ideal ones would be to allow in contact with either side of the surface such a velocity as the cooling or heating of the respective fluids might themselves generate when the body of the fluids were otherwise at rest, and these velocities are not only essentially indeterminate in any given case, but they would vary very greatly according to the position and shape of the surface. Thus, if the surface were a vertical smooth plane, the resulting velocities would be wholly different from what they would be if the smooth plane were fixed horizontally, and in the latter case the respective velocities would be wholly different according as the heating or the heated fluids were above the other. Each of these again would be entirely different from what they would be if the surfaces were cylindrical instead of plane, and if the surface were rough or irregular, the whole of the values would be further altered.

Thus, the number of different possible conditions which might be exactly defined, and all of which frequently occur in practical use, is infinite, and the number of different values of the co-efficient would have to be infinite also. If it were possible to obtain the mean velocity of a very thin film of either fluid nearest to the solid surface, even this would not determine accurately the temperature of the infinitely thin layer of the fluids in actual contact with the surface.

But even abandoning as practically unattainable the use of co-efficients based strictly on exactly defined conditions, there must be yet a further broad distinction between any co-efficients which could be used in practice and those which could be experimentally obtained under conditions as nearly approximating as possible to the scientific ideal.

The value of K for any given surface which is sought for in practice is a sort of bastard mean of the actual semi-scientific value of K, the mean being taken over the entire surface in question.

Take, for instance, the case of an ordinary radiator. Let it be assumed that the accurate value of k is known for all temperatures, and for all differences of temperature between the heating and the heated fluids, and that also in the radiator under consideration, the temperature both of the heating and heated fluids at each part of the surface is known. Then taking any elementary area of the radiator, whose value may be indicated as dS, assume that at this point of the surface the internal temperature is θ_i and the temperature of the air in immediate contact with the radiator is θ_o, then we have

$$dW = dS \times k \times (\theta_i - \theta_o)$$

Summing these elementary quantities over the entire surface of the radiator we have a total heat given off

$$= \Sigma\, dW = \Sigma\, dS \times k \times (\theta_i - \theta_o)$$

where k is the accurate and variable co-efficient.

Now the practical value of K which is sought for in experiments is such that

$$S \times K \times (t_m - t_a) = \Sigma\, dW = \Sigma\, (dS \times k \times (\theta_i - \theta_o))$$

or in other words the object is to find such a value for K that when it is multiplied by the difference between figures which are assumed to represent, respectively, the mean temperature of the radiator and that of the surrounding air, the result will be the mean emission of heat per square foot taken over the whole radiator.

In the case of the heat lost from a radiator, however, the value of each factor of this expression is open to very considerable doubt.

VALUE OF S.—Even about such an apparently simple matter as the actual surface of a radiator there may be, and is, considerable experimental doubt. Thus, if the same radiator were submitted to two experts for measurement in the most careful manner, it is found from experience that their measurement will vary as much as 5%, according to the method of measurement adopted.

VALUE OF t_a.—The *most convenient* value for t_a is obviously the temperature of the room, though, of course, this is not by any means the same thing as the mean temperature of the air surrounding the radiator, but it is the most convenient value for practical purposes and is therefore adopted.

MEAN TEMPERATURE OF A RADIATOR, t_m.—A water heated radiator has a varying flow and a varying return temperature. The temperature at different parts will vary over the entire surface. Each part of a radiator, in which a perfect circulation is maintained, should have a temperature somewhere between that of the flow and of the return. In other words, the return pipe should take off the coldest water in the entire radiator.

The value, which would be called the "mean temperature" of the radiator, in a scientific sense, would be determined as follows:—

Assume that the surface, having a total of S square feet, is divided up into a large number of small elements, each, let us say for example, .001 of a square foot in area. Assume that the actual mean temperature θ° of the surface of each one of these small areas is determined, and that they are so small that the temperature over any one does not vary appreciably from the mean temperature of that element.

Then the mean temperature of the radiator would be $\Sigma\, dS\, \theta / S$.

Of course, any practical process of obtaining the mean temperature according to this formula would be the work of many days. The time spent in making such measurements would be entirely wasted, seeing that in practice, not only does the distribution of temperature in different radiators vary, but that even in the same one it varies from hour to hour. Not only so, but this interpretation of the "mean temperature" is quite unconnected with present purposes.

The practical man on the other hand requires a definition such that the value of the "mean temperature" can be ascertained by means of thermometers easily and rapidly applied. There is an irreconcilable difference between these requirements.

If it were possible, by means of a circulator fixed within a radiator, to ensure that all parts were at exactly the same temperature, it would be possible to get with some accuracy figures which might be taken to represent not only the mean temperature of, but also the emission of heat from surface *at a uniform temperature* when massed together in that particular form, and placed in the particular conditions in which that radiator was actually placed in the laboratory, surrounded by still air of such and such a temperature, at such and such a distance from a wall, and so forth.

This might be a result of considerable scientific interest, but it would be of no practical value, because when such radiators are used in practice they are *not* provided with an internal circulator, their temperature is *not* uniform throughout from top to bottom, in all probability the radiators may not be of the same kind, and will *not* be fixed in the same way in a house or public building as they are actually fixed in the laboratory. The broad result would be that such an experimental result, obtained with never so great care and carried to a high degree of accuracy, would be of no practical use to a heating engineer, or, at least, would only be of use as a rough approximation. An experiment actually carried out with much less care, thought, and scientific accuracy, would be of a great deal more use, because its attendant conditions would probably approximate more closely to the conditions under which the radiator would be used in practice.

The real criteria of suitability of a formula for the mean temperature for practical use are—

(1) That it shall be a function of the inlet and outlet temperatures only, so that its value can be easily calculated.

(2) That the formula will give the same results for the mean temperature for all corresponding values of inlet and outlet temperature for which the radiator itself would give off the same amount of heat.

It is possible, for instance, as is frequently or generally done, to take the "mean temperature" of the radiator to be the mid-temperature between flow and return, or

$$\text{Mean temperature} = \frac{(t_f + t_r)}{2}$$

If this is a suitable expression of the mean temperature for present purposes, it follows that a radiator into which water is flowing, for instance, at 200°, and out of which it is flowing at 100°, is giving off the same total amount of heat as the same radiator, when the flow temperature is 160° and the return 140°, or the flow at 180° and the return at 120°, and so on.

If this is not true then any formula based on the mid-temperature must be wrong, that is, the formula will give the same result in cases when it ought, if it were correct, to give a different result.

It is obvious that this formula does not hold good in extreme cases. For instance, cases are quite common where the temperature of the flow pipe leading water into the radiator is high, say, 180°, and owing to bad circulation, the body of the radiator is practically cold, and the outflowing water nearly dead cold, say, 60°. It would be obviously incorrect to call the mean temperature of that radiator 120°, and to assume that the same amount of heat would be given off, as if it had a flow temperature of 130° and a return temperature of 110°.

Or, again, it is quite possible that one or more sections of a radiator might become blocked up by silt or deposit, or one or more of the sections might be short circuited from some other cause. Then, if the quantity of water flowing through the radiator were maintained constant, it is clear that the greater the number of sections short circuited the higher would the return temperature be, and any formula based exclusively on the temperatures would give a higher result for the heat emission according as the radiator itself was more inefficient.

Cases have been in fact observed in which a large part of the surface of a radiator has been many degrees below the temperature of the outlet, only a small part connecting the inlet and the outlet being really hot. Indeed, this phenomenon is observable to a certain degree in practically every radiator, especially in long radiators connected above and below on the same or opposite sides.

The above examples illustrate the case of a radiator within which the circulation is abnormally bad. Consider on the other hand the following example :—

Assume that a radiator is provided, as suggested on page 272, with an internal circulator, such that all parts are uniformly at a constant temperature. This fairly represents the case of perfect circulation within the radiator. The temperature is supposed to be maintained by an inflowing stream of hot water. That of the return water is the same as that of the body of the radiator.

The real mean temperature here is obviously that of the outflowing water, whereas the mean calculated on the mid-temperature basis, would be half way between the flow temperature and the actual mean, so that here again the mid-temperature formula is seen to be widely in error.

Of course, the above are extreme cases, but they are sufficient to show that the mid-temperature between flow and return, or, indeed, any formula based exclusively on the inlet and outlet temperatures, *and without some factor representing the character of the circulation within the radiator itself,* cannot be in all cases a reliable indication of the "mean temperature" of a radiator for present purposes, and that a calculation based on the mid-temperature which produces such large errors in extreme cases can hardly be supposed to be closely accurate in average cases.

The better the circulation within the radiator the more nearly does the real mean temperature approximate to that of the outlet. When there is no circulation within the radiator at all and no convection currents the mid-temperature is a fair approximation, as is proved below, for small differences only.

Now in actual radiators the internal circulation varies very much according to the type of connection adopted, and also to a considerable extent on the design of the radiator itself.

In practice it is necessary to have a value for the mean temperature which can be used by the practical man, and this value must of necessity omit all reference to the state of the circulation within the radiator, for this cannot be generally known, and the following investigation is directed to obtain a formula which is the most suitable one for general use, that is to say, is the least inaccurate, for it will be evident from the preceding that scientific accuracy is here incompatible with practical utility.

The assumptions that are made in the following investigation are that there is no circulation within the radiator itself, but that the water is regularly and uniformly displaced between the flow pipe and the return pipe, that is to say, that the water follows the same course as in the case of a horizontal pipe through which a stream of hot water is passing.

It is also assumed that the transmission per square foot from any part of the surface is proportional to the difference in temperature between the water within the pipe at that point and the air of the room. In other words, that the value of K is constant between the temperature of the flow and that of the return, so that for any part of the surface

$$dW = dS \times (\theta_i - \theta_o) \times K.$$

Note should be taken that each one of these assumptions is only approximately accurate, but they are the most practical assumptions to make, and their divergence from actual accuracy is probably not great enough to be of any serious moment.

Loss of Heat from a Pipe.

Assume for purposes of illustration that a stream of water equal to Q pounds per hour is flowing through a straight length of pipe l feet long, which is c feet outside circumference.

Let θ_1 be the difference between the temperature of the water in the pipe and the room air at the beginning, and θ_2 at the end of this run of pipe, the value of K being constant between the extreme temperatures, t_a being the air temperature and t_1 t_2 the temperatures of the water at the beginning and end of the pipe. Then $\theta_1 = t_1 - t_a$ and $\theta_2 = t_2 - t_a$.

Consider any infinitely short length, dx long, of this pipe at a distance x ft. from the return end. Let the difference of temperature between the water and the outside air be θ at this point, and the difference of temperature between the two ends of this short length be denoted by $d\theta$. Then the loss of heat per hour from this short length of pipe would be

$$c \times dx \times \theta \times K \text{ B.T.U. per hour.} \qquad \text{(I.)}$$

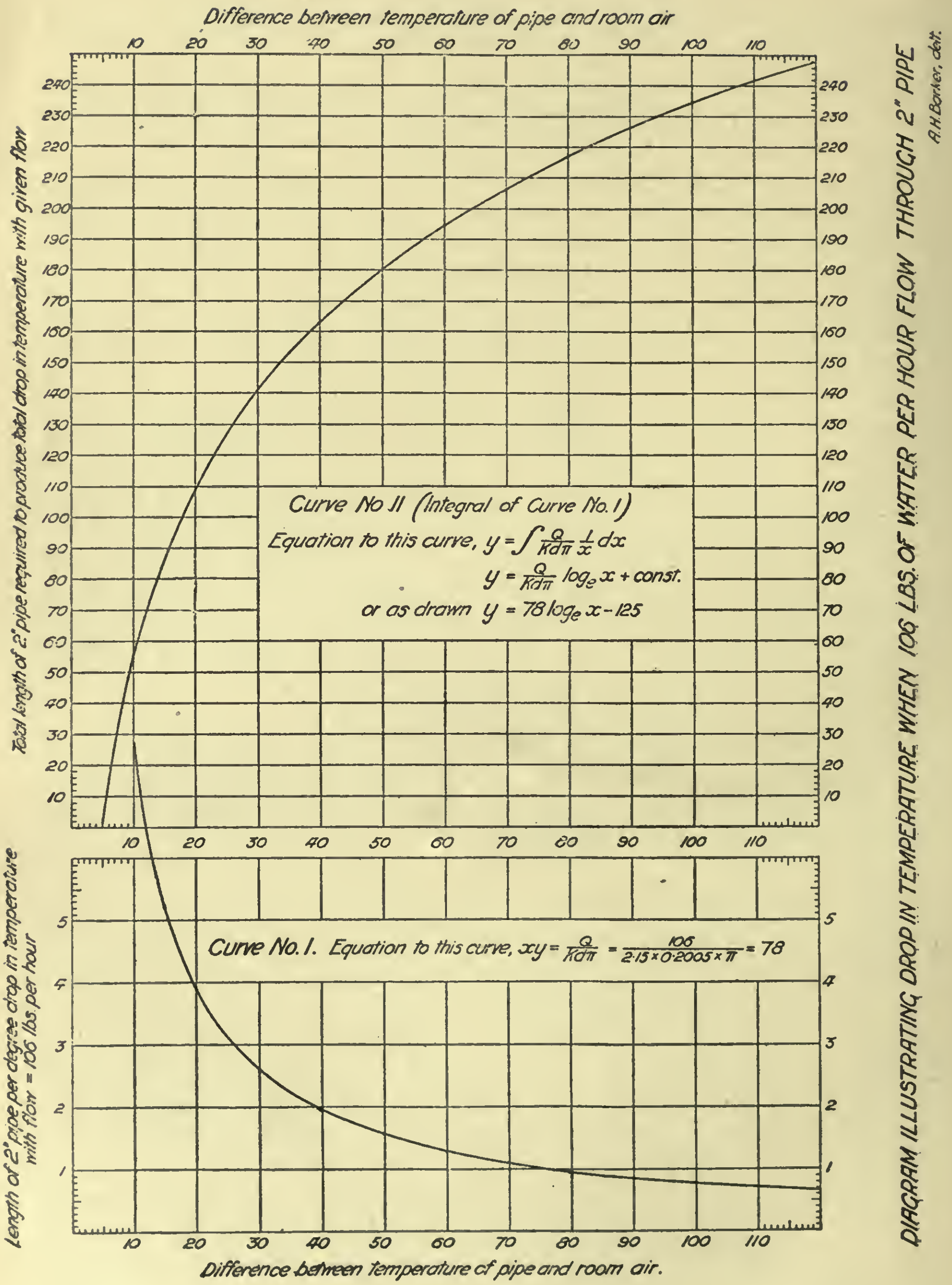
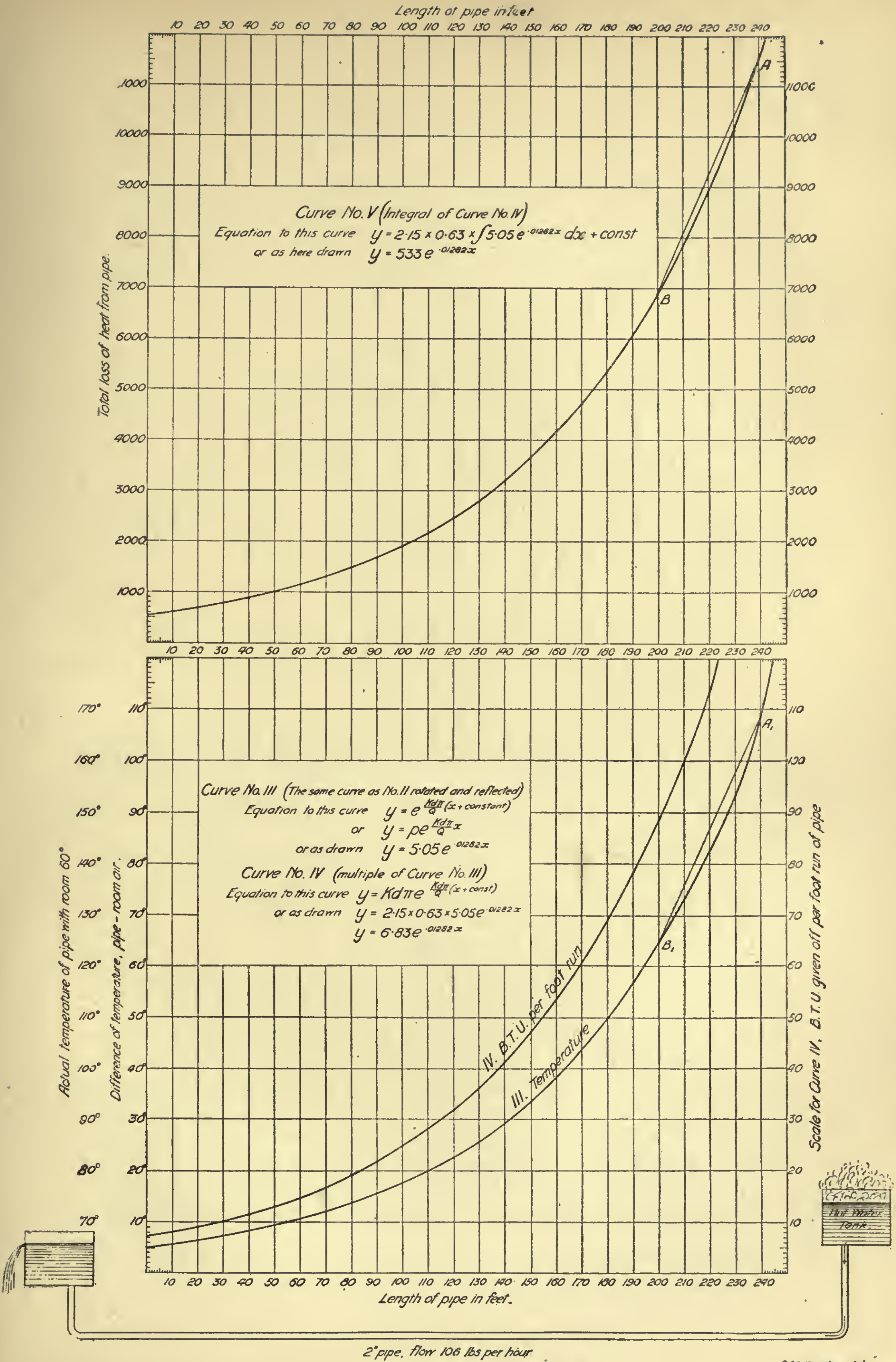


DIAGRAM ILLUSTRATING DROP IN TEMPERATURE WHEN 106 LBS. OF WATER PER HOUR FLOW THROUGH 2" PIPE
A.H. Barker, delt.
Difference between temperature of pipe and room air
10 20 30 40 50 60 70 80 90 100 110
Total length of 2" pipe required to produce total drop in temperature with given flow
10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200 210 220 230 240
Curve No II (Integral of Curve No. I)
Equation to this curve, $y = \int \frac{Q}{Kd\pi} \frac{1}{x} dx$
$y = \frac{Q}{Kd\pi} \log_e x + \text{const.}$
or as drawn $y = 78 \log_e x - 125$
Curve No. I. Equation to this curve, $xy = \frac{Q}{Kd\pi} = \frac{106}{2{\cdot}15 \times 0{\cdot}2005 \times \pi} = 78$
Length of 2" pipe per degree drop in temperature with flow = 106 lbs. per hour
1 2 3 4 5
Difference between temperature of pipe and room air.

FIG. 81.



A H Barker. del.

FIG. 82.

The loss of heat may also be represented by Q × $d\theta$. Hence

$$Q\,d\theta = c\,\theta\,K\,dx \qquad \text{(II.)}$$

or

$$\frac{dx}{d\theta} = \frac{Q}{c\,K} \times \frac{1}{\theta} \qquad \text{(III.)}$$

or

$$\frac{dx}{d\theta} \times \theta = \frac{Q}{c\,K}$$

Hence, integrating both sides of this equation

$$x = \frac{Q}{c\,K} \log \theta + \text{constant or } \frac{x\,cK}{Q} + \text{constant} = \log \theta \qquad \text{(IV.)}$$

or expressed in another way

$$\frac{d\theta}{\theta} = \frac{c\,K}{Q}\,dx$$

therefore

$$\int_{\theta_2}^{\theta_1} \frac{d\theta}{\theta} = \frac{c\,K}{Q} \int dx$$

$$\log_e \theta_1 - \log_e \theta_2 = \frac{c\,K}{Q} l \quad \text{or } \log \frac{\theta_1}{\theta_2} = \frac{c\,K}{Q}\,l$$

The real meaning of this result will be better understood by those unfamiliar with mathematical processes by considering the graphical interpretation of it.

Plot a curve (I.) exhibiting the relation between the value of θ as abscissa, and $\frac{dx}{d\theta}$ as ordinate. Or in other words, between the difference of temperature and the length of pipe which would give a fall of temperature of 1° under the above conditions.

The curves are drawn to scale for the particular case, size of pipe, 2″ wrought iron, $c = \frac{2.406\,\pi}{12} = 0.61$ feet. Quantity of water, 106 pounds per hour. Length of pipe, 250 feet. $t_1 = 180$. $K = 2.15$.

Equation III., above, shows this curve to be a rectangular hyperbola. The area under this curve must obviously represent the total length of the pipe. On integrating this curve graphically, a curve of logarithms (II.) is obtained. (For a full explanation of this, see the Author's work, "Graphical Calculus," Longmans & Co., 5th Edition.)

This shows that the length of pipe which will cause a given drop in temperature is proportional to the difference in the logarithms of the two temperature-differences, or, in other words, proportional to $\log \frac{\theta_1}{\theta_2}$

Interchange the abscissa and ordinate in this curve of logarithms. We thus get a curve (III.) of temperature differences plotted on the length of pipe as base.

From the meaning of a logarithm the equation to this curve is obviously

$$\theta = e^{\frac{x\,c\,K}{Q} + q} \quad \text{or } \theta = p\,e^{\frac{x\,c\,K}{Q}} \quad \text{where } p = e^{q}$$

This is the same equation as IV., on page 278, written in another way.

Now since the loss of heat from any element of this pipe is proportional to the difference of temperature θ, and to the length of the element, the total area under the curve must be proportional to the total loss of B.T.U. from the pipe. The heat emitted per foot run is given by curve IV., whose height is cK times the height of curve III.

The area of this curve is obtained by a second integration as follows :—

Loss of heat from any short length dx.

$$dH = c \times dx \times \theta \times K$$

$$= cK\,p\,e^{\frac{x\,c\,K}{Q}}\;dx$$

Hence

$$H = cK\,p \int e^{\frac{x\,c\,K}{Q}}\,dx = \frac{p\,(cK)^2}{Q}\,e^{\frac{x\,c\,K}{Q}} + \text{constant}$$

Now consider the state of temperature and the loss of heat from each portion of this pipe, as shown by the above equations and curves.

In order to illustrate the numerical meaning of the equations calculate what is the total loss of heat from the pipe between the points marked on Curves III., IV. and V., Fig. 82, as 200 and 240 feet, that is to say, for a length of 40 feet of 2″ pipe.

The temperature at each point is given by Curve III., whose equation is

$$y = 5.05\,e^{.01282\,x}$$

The temperature at the point marked 240 is given by

$$y = 5.05\,e^{.01282 \times 240} = 5.05\,e^{3.0768}$$

From a table of logarithms it may be found that

$$e^{3.0768} = 21.7$$

The temperature difference, therefore, between the room air and the water in the pipe at this point is 5.05 × 21.7 = 109.59.

Similarly the temperature difference at the point marked 200 on plan

$$= 5.05 \times e^{2.5640} = 5.05 \times 13.0$$

Whence the temperature difference is found to be 65.65.

Calculated on the ordinary formula, therefore, the mid-temperature difference is

$$\frac{109.59 + 65.65}{2} = 87.62$$

The actual heat lost, calculated mathematically, is proportional to the area under the *curve* A_1 B_1, on Curve III. The actual value of the heat lost may be calculated from Curve V., whose equation is

$$y = 533\ e^{.01282\ x}$$

For this curve, V., the value of y at point 240 is similarly found to be 11,566. The value at point b is found to be 6,929. The actual heat lost, therefore, is the difference between these figures, or 4,637 B.T.U. per hour.

If calculated on the ordinary mid-temperature formula the value is

$$2.15 \times 0.2005 \times \pi \times 40 \times 87.62 = 4{,}720 \text{ B.T.U.}$$

The total heat lost from the pipe between these two points is in this calculation assumed to be the area shown on Curve III., under the straight line A_1 B_1.

This example will give an idea of the degree of accuracy of the mid-temperature formula when applied to a pipe.

It will be seen that accurately the total loss of heat from 40 feet of this pipe is 4,637 B.T.U. per hour. It will also be seen that for a short length like 40 feet, the curve of fall of temperature is not exactly a straight line, but is not far off.

The error, therefore, in this case, in assuming that the mean temperature of the pipe is the mid-temperature, is small.

The matter may be considered in another aspect.

The total heat lost from the pipe is

$$Q\ (t_1 - t_2)$$

We require to obtain this loss in the form

$$l\ c \times \theta_m \times K$$

where θ_m is the mean value of the temperature difference, that is to say,

$$l\ c\ \theta_m\ K = Q\ (t_1 - t_2)$$

but Q is equal to

$$\frac{l\ c\ K}{\log.\ \dfrac{\theta_1}{\theta_2}} \quad \text{See equation IV. on p. 278.}$$

whence the actual value of the mean temperature difference is

$$\theta_m = \frac{t_1 - t_2}{\log.\ \dfrac{\theta_1}{\theta_2}} = \frac{43.94}{\log.\ 1.67} = \frac{43.94}{0.5128} = 86^\circ$$

as against 87.62 when calculated on the mid-temperature basis.

Fig. 83 shows the relation between the temperature in the same pipe as before when the quantity of water flowing varies from the amount 106 pounds per hour, for which Fig. 81 is calculated. It shows that the smaller the quantity of water the more rapid is the fall of temperature in the pipe.

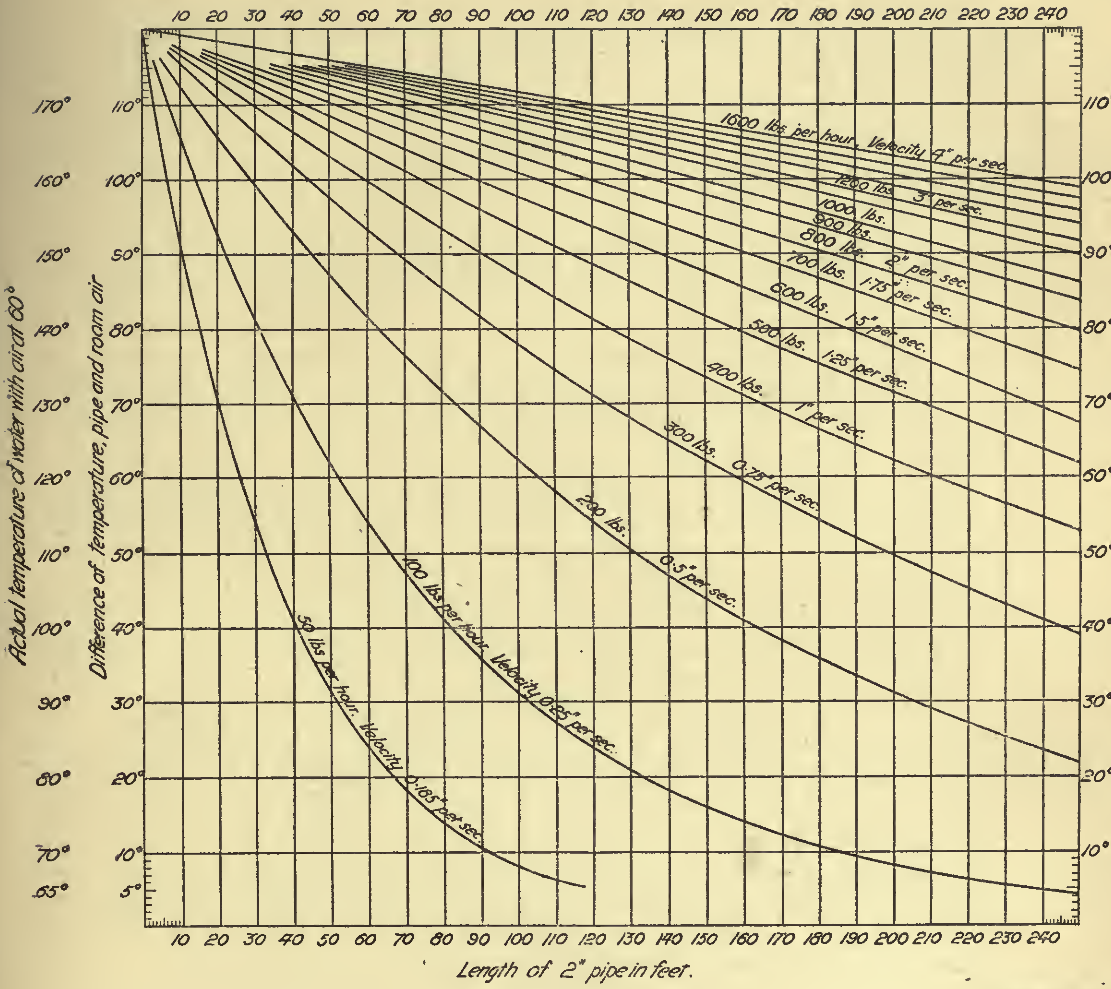


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FIG. 83.

Difference between Temperature of Radiator and Pipe.

Now the loss of heat from a radiator may be considered in exactly the same way, but there are obvious differences between the two cases of a radiator and a pipe.

The hot water enters the radiator as a rule, though not always, at the top, the return always leaves at the bottom.

It is the custom to treat a radiator as though it were in the same condition as a pipe. The essential difference between the two cases is due to the fact that convection currents are set up within the vertical radiator which cause the hotter water to accumulate near the top, so that there is not a regular fall of temperature between the top and the bottom as exists in the case of a pipe of moderate length through which a constant stream of hot water is flowing.

This effect of itself tends to increase the amount of heat given off to a degree which it is almost impossible to calculate without very great complications of symbols.

The distribution of temperature within a radiator varies according to the method of connection. If both connections are made at the bottom, as is the common practice in this country, a circulation of water is set up throughout the radiator. The first section acts as a rising pipe and the distant sections as falling pipes, just as in an ordinary hot water installation. The temperature throughout the radiator, when connections are made at the bottom, is therefore much more uniform than when the flow connection is made at the top.

In the latter case, if the radiator is supplied with an insufficient quantity of water, the result will be that the top part of the radiator will be hot and the lower part quite cold, whereas if the same quantity of water had been circulated to a radiator with bottom connections the temperatures would have been more nearly uniform over the entire radiator.

The result is that with the same quantity of water flowing through the radiator, the return pipe from a radiator having top and bottom connections would be colder than with a radiator connected at the bottom. Therefore the top connection would secure a better result from a given insufficient flow of water, than would a bottom connection.

The condition as to temperature of a radiator having top and bottom connections may be seen from diagram, Fig. 84, which is plotted on a vertical base line showing in its horizontal ordinate the temperature at all points of the radiator, and the calculated temperature in a horizontal pipe on the principle explained in connection with Fig. 81.

This pipe is supposed to be of such a length that it would abstract from the water exactly the same quantity of heat as does the radiator, or in other words, that the total emission of the pipe should be the same as that of the radiator. The scale of the vertical base line for the pipe curve is assumed adjusted to correspond to the height of the radiator.

In these circumstances it will be seen that the temperature at the top half of the radiator is greater than the corresponding temperature in the pipe, and at the bottom half lower. But the area under the radiator curve which represents the total heat emission must be the same as that under the pipe curve. As the quantity of water flowing through the radiator increases from an insufficient minimum, the temperature adjusts itself in a manner which will be understood by an inspection of the successive curves.

It is, however, the custom to calculate the emission from a radiator as though it were under exactly the same conditions as a horizontal pipe in respect of its temperature. It will be seen from a consideration of these diagrams that this principle is essentially incorrect. The formula to give the value of the "mean temperature" of a radiator is one that can only be determined empirically and cannot be the mid-temperature. As explained on page 273, that formula should be such that the radiator will emit the same amount of heat for all conditions of flow and return temperature for which the formula gives the same value for the mean temperature. The mid-temperature basis does not satisfy this condition, and is therefore necessarily incorrect. This question needs experimental investigation.

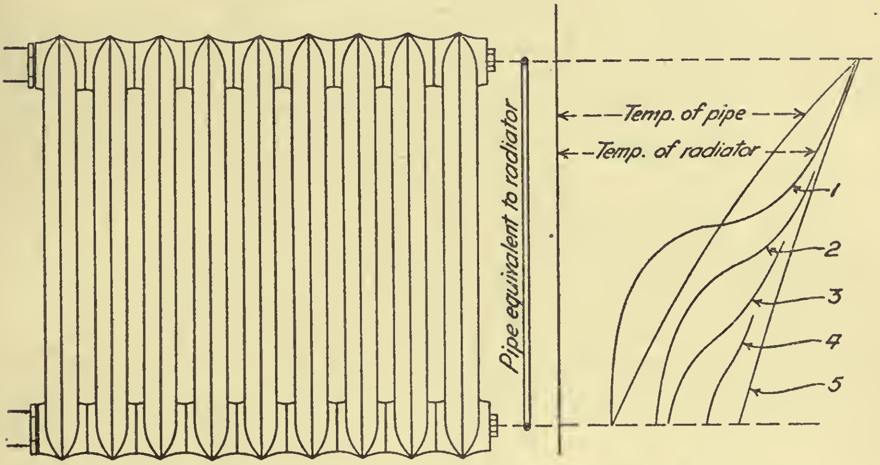


Diagram illustrating comparison of temperature of water in a pipe and in a radiator connected above and below when flow insufficient.

FIG. 84.

Careful note should be taken of the fact that with a given flow through a radiator, the worse the circulation is within the radiator, that is, the greater the difference in temperature between top and bottom, the greater is the total emission of heat, and that for this reason the method of connection of radiators above and below, causing, as it does, less movement of the water within the radiator, is a more efficient method than bottom connections.

It will be seen by an inspection of the curve No. 5 on Fig. 84 throughout its whole length, that provided the difference of temperature is relatively small, no very substantial error is caused by taking the mid-temperature as the mean temperature for a horizontal pipe, but that if the difference of temperature is great, so that the lower temperature difference is not very far from zero, the mid-temperature is widely in error.

The determination of the practical value of K then is made on the assumption that the formula

$$W = s \times (t_m - t_a) \times K$$

is accurate. Careful note must be taken of the assumptions which this formula makes, namely,

(1) The loss of heat is proportional (*a*) to the temperature, (*b*) to the area of surface.

(2) That the value for t_m, namely, $\frac{t_f + t_r}{2}$ is accurate for the ranges of temperature within which the formula is applicable. In other words, that a radiator will give off the same amount of heat to the surrounding air in all cases in which the value of $\frac{t_f + t_r}{2}$ gives the same value.

None of these assumptions is in fact correct. When a large amount of surface is massed together, as in the case of a radiator, the heat emission is not proportional to the surface, because the temperature in the neighbourhood of the radiator is increased, so that the value of the room temperature is not a proper value to take for t_a.

It has been already shown that the value $\frac{t_f + t_r}{2}$ is not a correct interpretation of t_m.

In so far as these two assumptions therefore are in error, they are compensated for in practice by giving different values of K for different cases.

It is not found necessary to give different values for K for every corresponding value of inlet and outlet temperature, because the mid-temperature is found to be sufficiently near to accuracy for practical purposes. Different values are, however, given for K

(1) For different classes and arrangement of surface.
(2) For different temperatures.
(3) For different quantities of surfaces massed together.

To Determine the Value of *K*.

The method of experiment is, of course, to obtain the total emission from different kinds of radiating surface as a whole, by experimental observation, and determine the corresponding values of K by division for each case.

This, of course, leads to a large number of different values being found of K. The value of K is determined in such a way as to mask the mistakes in the fundamental theory. In fact, the object of these experiments is to obtain values for K, such that the formula

$$W = S \times (t_m - t_a) \times K$$

will be correct in practice, and different values of K are necessary not only for different arrangements and qualities of surface, but also for different magnitudes of the same kind or arrangement of surface, or for different ranges of temperature. For this is the most convenient formula in which to calculate the heat emission of a radiator.

If it be assumed that the heat emission is a function of the mean temperature, a mistake in the determination of the value of that mean temperature produces a necessary mistake in the calculations, unless different values of K are also given for every difference in either inlet or outlet temperature, or both, which would give an even more unmanageable number of values than those which are found necessary at present.

The whole process of increasing the number of different values for the co-efficients is an entirely artificial method of concealing the inaccuracy of the theory, and the whole formula and theory is made correct for practical cases by determining appropriate values of K for a large number of different cases.

From some extensive experiments made by the Author, which are not yet complete, it would appear that the value which expresses better the average or mean temperature of a radiator than does the mid-temperature is given by the following formula :—

For radiators connected below $\dfrac{t_f + 2\,t_r}{3}$

For radiators with very good flow connected above and below $\dfrac{2\,t_f + t_r}{3}$

But in the very careful and elaborate series of experiments made with the unequalled resources of the Technical High School, at Charlottenburg, by Rietschel, which are probably among the best and most complete experiments which have yet been made, the basis of the mid-temperature was assumed as the mean temperature of the radiating surface, and all co-efficients are evaluated accordingly. This assumption being probably based on the mathematical investigation on page 278, and one cannot afford to dispense with these valuable experimental results on the ground that the basis on which they are calculated may be to some extent unsound. If a more satisfactory expression for the mean temperature were adopted it would be necessary to have values for the co-efficient K evaluated afresh.

Of course, the accurate value of the total emission only becomes practically important when the temperatures are high, that is to say, when the conditions are near to those which ordinary guarantees in contracts are intended to apply, and those with which the experiments were made. Within this comparatively limited range, calculations made on the basis of mid-temperature cannot be greatly in error as applied to the only conditions for which accuracy is of importance.

In practice the values of the difference in temperature between flow and return in a well fixed radiator to which the circulation is good, lie between 20° and 60° as outside limits. A radiator whose temperature differences are outside these limits would be regarded as suffering from short circuit or bad circulation. To the latter calculations based on the mid-temperature cannot apply.

Transmission of Heat from Liquid to Liquid.

It has been already explained that the value of the transmission coefficient K from liquid to liquid through a metal plate varies largely with the velocities of the liquids. When the liquid is in regular motion over the surface, it is clear that as it flows it is parting with its heat, or gaining heat, as the case may be, and therefore it falls in temperature from point to point, and it is impossible to take any one particular temperature as being the temperature of the liquid.

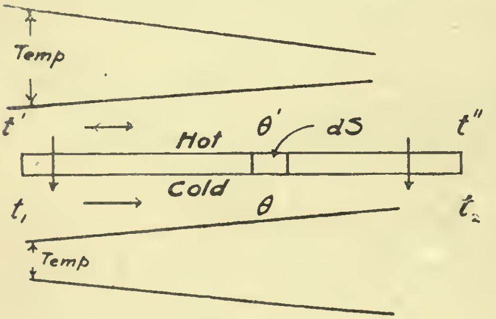


FIG. 85.

It is therefore advisable to analyse the loss of heat as far as possible. If the warmer liquid is flowing in one direction, and the cooler liquid in the opposite direction, the conditions are evidently different from what they are when both liquids are flowing in the same direction, and both these conditions are different from the state of things when neither liquid has any considerable velocity in contact with the solid.

The following treatment is due to Rietschel :—

Suppose a plate of metal, Fig. 85, has liquid on each side of it, moving in the same direction. The warmer liquid becomes cooler as it travels, and the cooler liquid becomes warmed, and suppose that a quantity of heat W is transferred through the plate per hour.

Consider an elementary surface dS, having temperatures on either side of it $\theta' \theta$. Clearly we have

$$dW = dS\, K\, (\theta' - \theta)$$

Also

$$\frac{dW}{W} = \frac{d\theta}{t_2 - t_1}$$

Whence

$$dS = \frac{d\,\theta\, W}{K\,(\theta' - \theta)\,(t_2 - t_1)}$$

In order to solve this equation it is necessary to eliminate $\theta' - \theta$.

Now the whole reduction of temperature of the hotter fluid, before reaching dS, must bear the same relation to the gain in heat of the cooler fluid, as the whole loss of temperature in the hotter fluid does to the whole gain of temperature in the cooler fluid. That is

$$\frac{(t' - \theta')}{\theta - t_1} = \frac{t' - t''}{t_2 - t_1} = \frac{Q_2}{Q_1}$$

because the quantity of water flowing over the first section is the same as over the remainder.

Adding (1) to each side of this equation, we have

$$\frac{t' - \theta' + \theta - t_1}{\theta - t_1} = \frac{t' - t'' + t_2 - t_1}{t_2 - t_1}$$

If then we put

$$t' - t_1 = \triangle$$

and

$$t'' - t_2 = \triangle_1$$

the above becomes

$$(\theta' - \theta)\,(t_2 - t_1) = (\triangle_1 - \triangle)\,\theta - \triangle_1\, t_1 + \triangle\, t_2$$

If the above value is inserted in the equation for dS above, we get

$$dS = \frac{d\,\theta\, W}{K\,\left\{(\triangle_1 - \triangle)\,\theta - \triangle_1\right\}\, t_1 + \triangle\, t_2}$$

which on integration between the limits t_2 and t_1

$$S = \frac{W}{K\,(t' - t_1 - t'' + t_2)} \log_e \frac{t' - t_1}{t'' - t_2}$$

An exactly similar treatment for the case of liquids flowing in opposite directions gives the equation

$$F = \frac{W}{K\,(t' - t_2 - t'' + t_1)} \log \frac{t' - t_2}{t'' - t_1}$$

If it is assumed that the warmer liquid is at rest, that is to say, there is no regular velocity, and that it has throughout the temperature t, that is, $t' = t'' = t$, the above equation becomes

$$F = \frac{W}{K\,(t_2 - t_1)} \log \frac{t - t_1}{t - t_2}$$

If the temperature differences between the hotter and the colder liquid are not very great, and if also there is no definitely maintained velocity, both equations become

$$F = \frac{W}{K \left(\dfrac{t' + t''}{2} - \dfrac{t_1 + t_2}{2} \right)}$$

and if both liquids are at rest, and have the same permanent temperatures, the equation may be put as

$$F = \frac{W}{K\,(t - t_0)}$$

The above treatment applies only to cases where a definite velocity of the different streams can be relied upon. This is very rarely found in practice, and though the principle is important, its direct numerical application to practical work is somewhat limited.

Quantities of Water Required.

It will be seen from the previously explained facts that the question of the satisfactory working of a hot water heating apparatus is entirely that of producing an adequate flow of hot water through the radiators in the system.

It will be well, therefore, to show the quantities of water which will pass through different sizes of pipe at different velocities, and also to show how to calculate the quantity of water which must flow through a radiator in order that the necessary temperature may be maintained.

The internal cross sectional area of any pipe, whose diameter is d, is given by

$$a = \frac{\pi\, d^2}{4}$$

and the quantity of water delivered through such a pipe at a mean velocity v'' per second is given by

$$Q = \frac{\pi\, d^2\, v}{4} \text{ in cubic inches per second.}$$

If the diameters of pipe are given in feet and the velocity of the water in feet per second, the same formula will give the quantity of water in cubic feet per second.

The weight of water per second depends, of course, on the density of the water or weight in pounds per cubic foot. This varies according to the temperature.

Let D be the density in pounds per cubic foot, then the number of cubic inches in one pound is equal to

$$\frac{1{,}728}{D}$$

Hence, the formula for the weight in pounds of water delivered by any pipe at velocity v'' per second is

$$\frac{\dfrac{\pi\, d^2\, v}{4}}{\dfrac{1{,}728}{D}} = \frac{\pi\, d^2\, v\, D}{4 \times 1{,}728} = \frac{\pi\, d^2\, v\, D}{6{,}912}$$

Similarly, if the quantities are given in feet, the formula is

$$\frac{\pi\, d^2\, v\, D}{4} \text{ pounds per second.}$$

These values are calculated and set out on the diagrams, Figs. 86 and 87, from which the quantity of water passing through any size of pipe at any mean velocity can be determined.

Quantity of Water Required by Radiating Surface.

If a quantity of water Q pounds per hour at a temperature t_1 passes into a radiator, pipe, or other radiating surface, and an equal quantity flows out of the pipe at the outlet, the amount of heat given up by the water in this passage through the radiator is

$$Q\,(t_1 - t_r) \text{ B.T.U.}$$

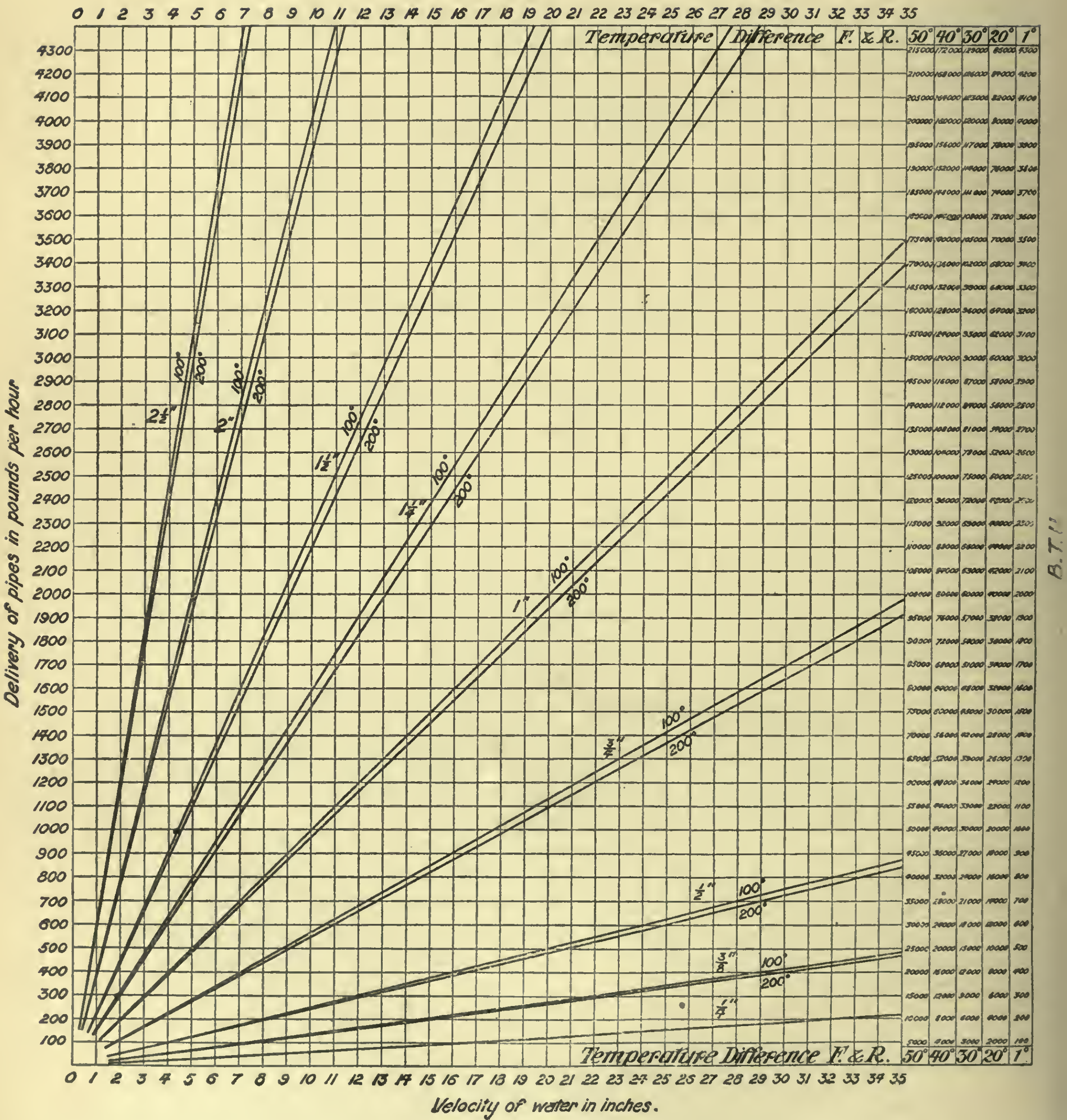
Assume this radiating surface has a surface S square feet, and has a co-efficient of conductivity K. Then the amount of heat is also given, assuming the mid-temperature to represent accurately the mean temperature, by

$$S \times K \left(\frac{t_1 + t_r}{2} - t_a \right)$$

If the value of K has been correctly determined it is obvious these two quantities of heat must be the same, that is to say,

$$Q\,(t_1 - t_r) = S\,K \left(\frac{t_1 + t_r}{2} - t_a \right)$$

From this equation any question relating to the flow of water through radiators can be solved, provided sufficient particulars are given.



CARRYING CAPACITIES OF WATER PIPES IN POUNDS AND B.T.U. PER HOUR AT ALL VELOCITIES FROM 0" TO 35" PER SECOND.

FIG. 86.

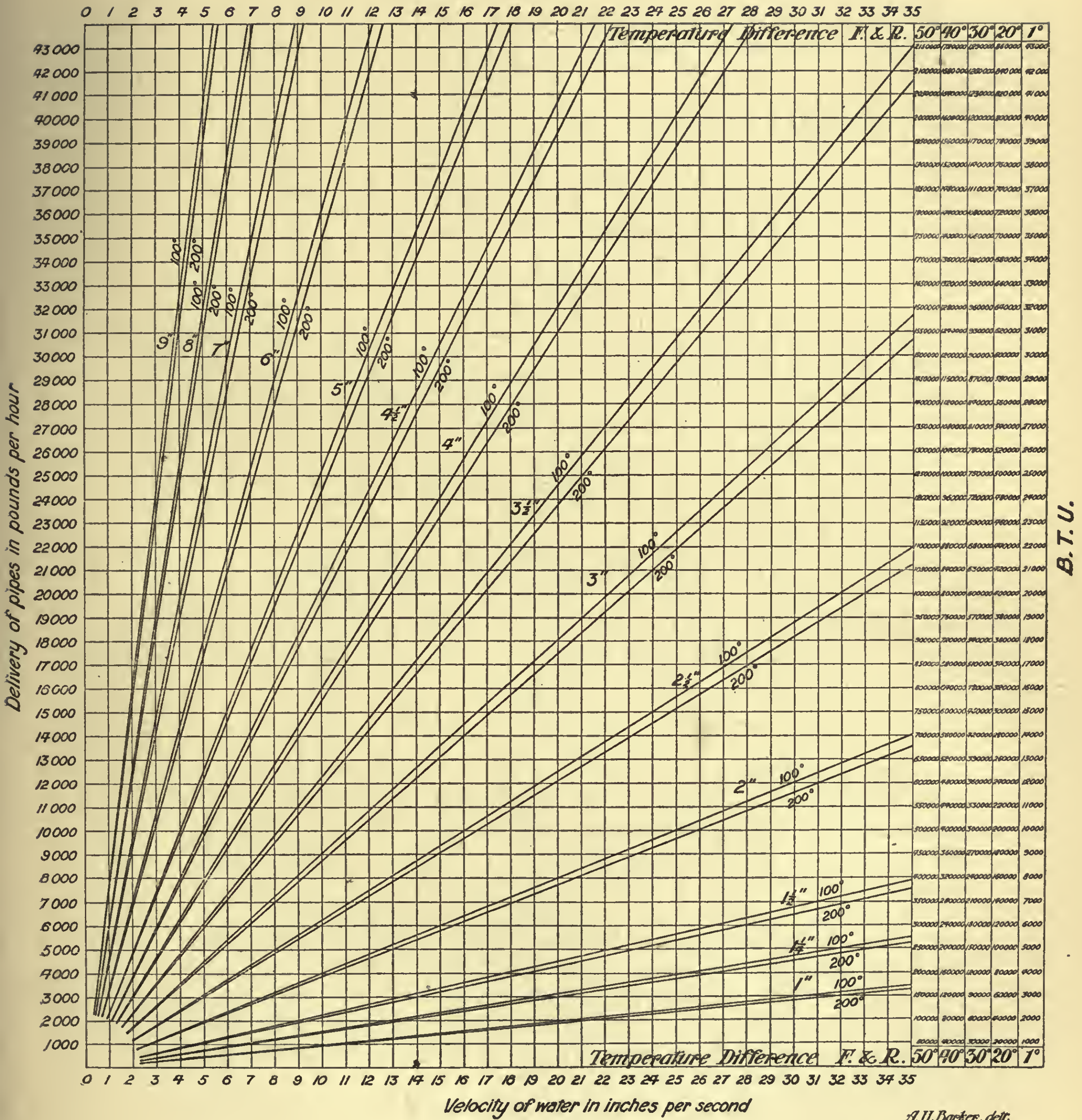


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FIG. 87.

Thus, if t_f and t_r are given and also S and K, the quantity of water is

$$Q = \frac{S\,K\left(\dfrac{t_f + t_r}{2} - t_a\right)}{t_f - t_r}$$

If $Q\ S\ K$ and t_f are given, t_r has the following value :—

$$t_r = \frac{2\,Q\,t_f - S\,K\,(t_f - 2\,t_a)}{2\,Q + S\,K}$$

If $Q\ S\ K$ and t_r are given, t_f has the following value :—

$$t_f = \frac{2\,Q\,t_r + S\,K\,(t_r - 2\,t_a)}{2\,Q - S\,K}$$

It is not practicable, however, to determine t_a from this formula, because a relatively large variation in t_a only makes a comparatively small variation in the other quantities involved, and the degree of accuracy with which the respective values can be determined is not sufficiently great to determine closely the value of t_a.

Take as an example the determination of t_r in the following circumstances :—

Let Q be 360 pounds per hour, and let S be 45 square feet, $t_f = 175°$, $t_a = 60°$, and K 2.05. Then the value of t_r will be as follows :—

$$\frac{2 \times 360 \times 175° - 45 \times 2.05 \times (175 - 120)}{2 \times 360 + 45 \times 2.05} = \frac{126{,}000 - 5{,}374}{720 + 92.25}$$

$$= \frac{120{,}626}{812.25} = 147°$$

Fig. 88 shows the amount of loss of heat per square foot of radiator surface for varying differences of temperature between flow and return calculated from the above formulæ, and also the quantity of water required per square foot in different circumstances. These will save a great deal of calculation.

It is to be noticed that when water at t_f passes into a radiator, the same *weight* of cooler water must of course pass out of the return outlet in the same time, but the return water being cooler than the inlet water its volume is less, and therefore the actual velocity of the water passing out of the return is slightly below that flowing into the flow. The difference, however, is not sufficiently great to cause any trouble in practical calculations.

It is customary to take the mid temperature of the water for the mean, for purposes of calculating velocities.

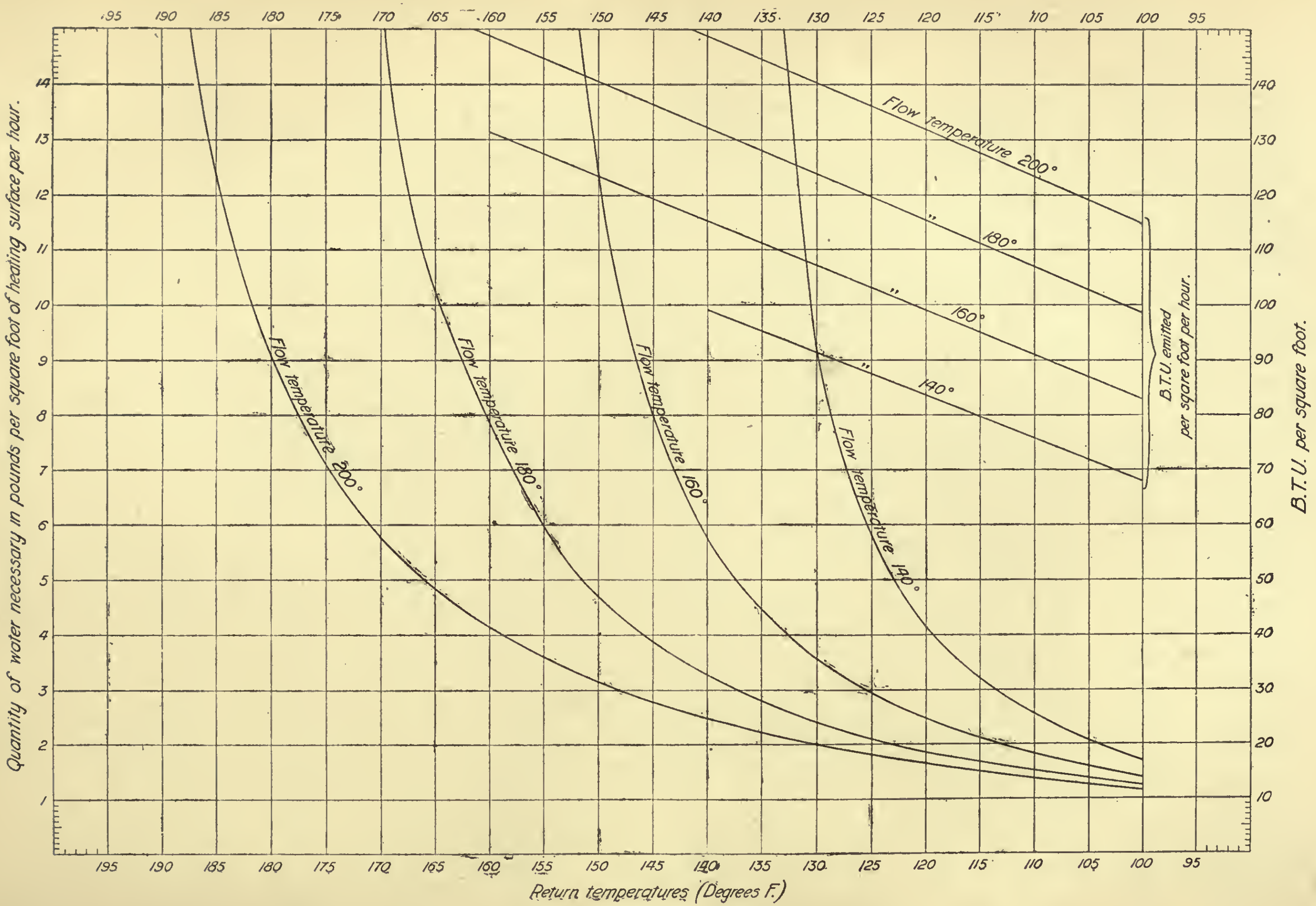


DIAGRAM ILLUSTRATING RELATION BETWEEN THE QUANTITY OF WATER REQUIRED FOR A RADIATOR AND B.T.U. EMITTED.

A.H.Barker, delt.

FIG. 88.

CHAPTER XIV.

DETERMINATION OF THE HEAT LOST FROM A ROOM IN PRACTICE.

PRACTICAL calculations of the heat lost from a room are made on the basis of the principles explained in the preceding chapter. The total surface of window glass, walls, ceilings, and floors is calculated and the heat lost through each separately estimated, the results added together. The quantity of air to be passed through the room for ventilation is then calculated and the heat necessary to raise its temperature to the given degree is added to the heat lost by conduction. To this sum is added a certain percentage, according to the position of the room or building.

In applying these theoretical principles, however, to practical cases it is very desirable for the engineer to obtain an exact conception of the relation which these bear to the facts of practical work. These realities are apt to be lost sight of if too close attention is paid to the theory of the subject.

In the development of the theory it is, of course, necessary to assume a constant temperature within the room to be heated, a constant and known co-efficient of conductivity, walls of definite material and thickness, uniform atmospheric conditions outside, and a definite quantity of air passing into and out of the room per hour. Without these assumptions it would be impossible to make any calculations at all.

It is further necessary to assume that the heating apparatus has been at work under these constant conditions for so long a time that a constant flow of heat is taking place from the walls, and that the walls have arrived at their final or ultimate temperature. It is only necessary to consider what the actual conditions are under which practically all heating apparatus are worked to see how far removed these are from the ideal ones which the theory assumes.

Variations in the Character of the Walls, etc., of the Building.

It is only rarely that the engineer can obtain any precise knowledge of the constructional character of the building which he has to heat. Not infrequently he has to work from plans on which only the roughest

FORM FOR CALCULATING EXACT VALUE OF HEAT LOST.

Name of Building. Architect.

Room.						Cooling Surface								Thickness of wall in inches	Temp. Fahrenheit			Co-efficients	Amount of heat			Supplement		Total B.T.U	Ventilation					Radiation supplied					Remarks
No.	Name.	Length	Width	Height	Cubic contents	Name	Aspect	Length	Height	Surface	Number	Deduct	Calculate		Inside	Outside	Difference		Loss	Gain	Total	Considering aspects	For interruption of heating	Loss of heat Units per hour	Vol. of air per hour	Rise of temp.	Co-efficient	B.T.U. for air interchange	Total for exposure and air	Kind	Quantity	Mean temperature difference	Co-efficient	B.T.U.	

FORM FOR APPROXIMATE CALCULATION OF HEAT LOSS FROM BUILDINGS.

No.	NAME	Length	Width	Height	Contents	COOLING SURFACES						Exposure factor	Temp. diff.	Heat lost	Kind	Quality	Factor	Equiv. of lin. pipe	Temp. diff.	Total B.T.U.	REMARKS
						Kind	Length	Width or height	Surface	Co-efficient	Equivalent Glass										

outline of the buildings, walls, roofs, etc., is shown. He has often, or perhaps in the case of a new building, generally, to make his estimate and to give a guarantee of results before many of the essential particulars, which have a large effect on the heat-losing properties of the building, have been determined by the architect. He can only get a general idea of the thickness of the walls from the plans, the material is frequently unknown to him.

Variations in the Heat Lost by Conduction.

It is known that the thermal conductivity of any particular brick varies considerably according to the magnitude of the pressure exerted on it during the manufacture. The more compact the brick the greater is its conductivity and the worse, therefore, it is from a heat-losing point of view.

The conductivity of the wall itself depends (1) on its condition as to dryness, (2) on the closeness with which the bricks are built together, (3) on whether an air space has been left between adjacent bricks, (4) on the nature and thickness of the mortar used and the plaster coating, (5) even the quality of the paper which is used in decorating the room has a considerable influence on the heat retaining properties of the wall.

It is generally impossible for the engineer to ascertain such particulars as the thickness of the glass which is to be used for the windows. It is always impossible to take into account the character of the workmanship, or the quality or condition of the wood of which the window frames are composed. The actual construction of the ceiling—a very important point —is frequently wholly unknown, similarly with the floor, and indeed every part of the apartment.

It will be seen from the accompanying table to what a great extent these important particulars exercise an influence on the heat losing properties of a room.

As regards the external conditions surrounding a building, it is customary to assume in framing guarantees, that the external air is at rest and dry. The actual conditions vary in practice from hour to hour. The velocity of the wind and the amount of moisture in the air both exercise a very important influence on the loss of heat by conduction, which it is hardly possible to calculate.

Variations in the Heat Lost by Interchange of Air.

The above remarks refer only to one portion of the heat loss.

Another portion is yet more indeterminate. The amount of air which passes at all times through a room with walls and windows of ordinary construction, by what is called "the natural interchange of air" is a quantity which passes the wit of man to calculate. This quantity changes every time the velocity of the wind impinging on the building alters. At one moment the interchange may be three times per hour. A change of wind half an hour later may reduce the quantity to ¾ of one interchange, and an hour later still the quantity may be two interchanges per hour.

The quantity of air passing in depends to a very large extent on the same conditions as the heat loss by conduction, viz., on the quality of the woodwork and fittings which are used in the construction, on the closeness of fit of the window frames, etc., and on the compactness and quality of the brickwork, on the quality and thickness of the plaster, on the nature of the wall paper used for decoration, and on the quality and constructional details of the floor and ceiling.

It will easily be seen that these two components of the heat loss are not and cannot be considered as independent of one another, although it is the custom to calculate them separately.

Owing to external wind pressures a certain varying volume of air always enters a room through the substance of the wall. A certain quantity of heat passes out of the same wall by conduction in the opposite direction. It is easy to see that some of this heat in its outward passage must be taken up and brought back into the room by the air on its inward passage, exactly how much it is impossible to say. This practical condition, which is, of course, of regular occurrence, must entirely alter the conditions under which heat passes from a room from those which theory assumes.

Influence of the Hygrometric Condition of the Air.

It has been pointed out in Chapter IV. that the hygrometric condition of the air must necessarily have considerable effect on the heat lost, not only because the specific heat of moist air is different from that of dry air, but also because the presence of moisture in the air produces an alteration, not only in the absolute conductivity of the walls, but also of the passage of air through them, and also on the amount of heat emitted from radiators or other heating surface. It is quite unknown what is the magnitude of these various alterations.

As, however, the hygrometric condition of the atmosphere is constantly changing from hour to hour, it would be impossible to take account of these ever varying changes, even if the degree of their influences were known, which it is not. In any case the proportional difference between the specific heat of moist air and dry air is not comparable with the proportionate probable error in the estimated quantity of air, and therefore to complicate the necessary calculations by introducing considerations of hygrometric condition would be waste of time.

Further, the weight of a certain volume of dry air is greater than that of the same volume of moist air, therefore any error due to neglect of a consideration of the hygrometric condition is on the side of safety so far as the ventilation loss is concerned.

It will be easily understood from the above analysis that there are so many factors having an important influence on the actual loss of heat which cannot be taken account of in practical calculations, that it is impossible to expect any exact accuracy in the estimate of heat losses. A large error may and often does arise, even although the best available co-efficients are correctly employed in the calculations.

If the combination of values which have been given for the co-efficient K, both as applied to walls of buildings and radiating surfaces, are suitable for practical use, it means, therefore, not that they are essentially accurate in a scientific sense, but that the combination has been found by experience to be satisfactory when applied in practice.

It is a mere delusion to assume that these values, if satisfactory in practice, can represent the amount of heat which would be transmitted under the assumed ideal conditions. It is probable that the amounts of heat which are assumed to be given off from radiating surfaces are more accurate than those assumed to be transmitted through wall surfaces and the like, because the former are based on laboratory experiments, conducted under conditions which enable some approach to reasonable accuracy to be obtained. The latter, in so far as they are satisfactory in practice, are and must of necessity be based on the experience of the application of the former to practical cases. They cannot be usefully based on experiments of the laboratory order for reasons already stated. The most that can be claimed for these co-efficients in respect of their theoretical accuracy is that they represent in a just proportion the relative cooling influences of the various parts of the bounding surface of a room. Their absolute values should depend on the experience of any given climate. They should be by no means necessarily constant for different climates per degree difference of temperature.

What, therefore, is really required by the heating engineer for practical purposes is not scientific co-efficients which shall show accurately the amount of heat in thermal units which would be transmitted through glass or homogeneous walls, when a constant temperature and uniform conditions are maintained on both sides. He should be able to calculate in regard to each room what may be called "a co-efficient of exposure," which will enable all rooms for which calculation is made to be arranged in proportion in the order of heat loss in such a way that it will be known that of two rooms of entirely different size and shape, if their "exposure factors," as calculated, are in the ratio, say, for example, of 3 to 2, then given identical weather conditions the amount of heat to be furnished to one would bear this proportion to the amount of heat which would have to be furnished to the other in order to maintain approximately the same temperature in both cases.

The engineer also requires what may be called "a weather factor" for the situation of the room, which will enable the severity of the climate in various situations and in various parts of the country to be known in their proper proportion, such that one would be able to say of a certain situation in a certain district in Scotland, that the weather conditions to be provided against in that locality would bear the ratio to the corresponding conditions in a certain locality, say in England, of say 5 to 4.

If this were done it would then be known that if two exactly similar and similarly constructed rooms were erected in the two localities, the respective amounts of heat which would be necessary to produce the same temperature in the two rooms would be in the same proportion as the respective weather factors.

What we have called the "weather factor" may however vary very considerably in the same locality, according to the exposure.

It would appear at first sight that the necessary figures could be more readily obtained by analysis of careful experiment on existing rooms rather than by careful calculations.

The guarantee conditions in this country, however, generally specify an external temperature of 30°. It is only once or twice a year that such a temperature is maintained in this climate for a sufficiently prolonged period to enable any reliable tests to be arranged for and carried out, and it is only necessary to consider the conditions in which such tests are generally made to see that it is almost as difficult to make a satisfactory test from which any definite information can be derived as it is to make accurate calculations.

Difficulties of Testing.

It has been explained that the entire theory governing the temperatures attained in buildings assumes that the heat is lost from the building under the conditions of a constant inside and outside temperature, a constant wall temperature, and a constant temperature of the heating medium within the radiator, and that in practice none of these conditions are even approximately fulfilled.

Variations of Outside Temperature.

In the British climate it is the rarest possible occurrence for the temperature to remain even for a single day sufficiently constant to enable a test to be taken, which by any stretch of imagination could be said to be a test of the accuracy of any calculations.

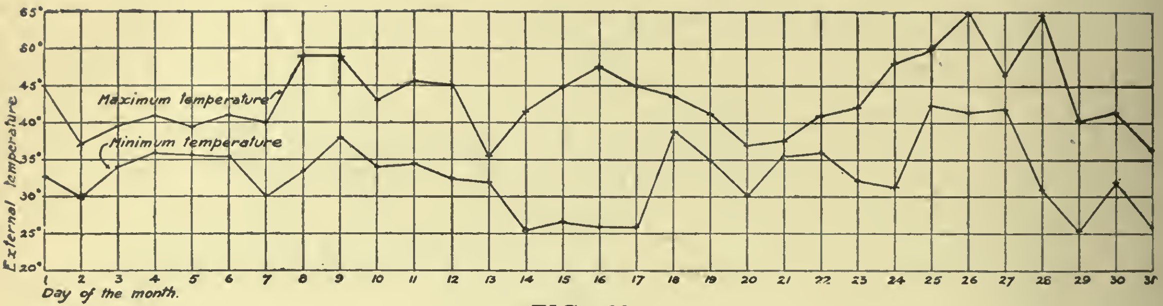


FIG. 89.

Fig. 89 illustrates the way in which the maximum and minimum temperatures vary during a normal month in this climate. The figures from which this diagram is plotted are taken at random from the Meteorological Office daily readings, and represent the actual readings taken in Cambridge during January, 1911.

In the great majority of whole periods of 24 hours, the temperature varies up and down within wide limits.

Fig. 90 also represents figures taken at random from the same source and shows representative temperature charts for two whole periods of 24 hours and the average for 35 years.

For that reason alone it is and must always be very difficult, if not impossible, to obtain a thoroughly satisfactory experimental verification on a large practical scale of the theory on which the calculations are founded.

For practical purposes the accuracy or otherwise of the calculations can only be judged by observing the internal and external temperature and the temperatures of the water throughout a long period of time for a large number of rooms. If in the main such temperatures are lower than desired then the values of the co-efficients K for walls and windows have been too low. If on the other hand the temperatures are higher than necessary the co-efficients adopted have been too high.

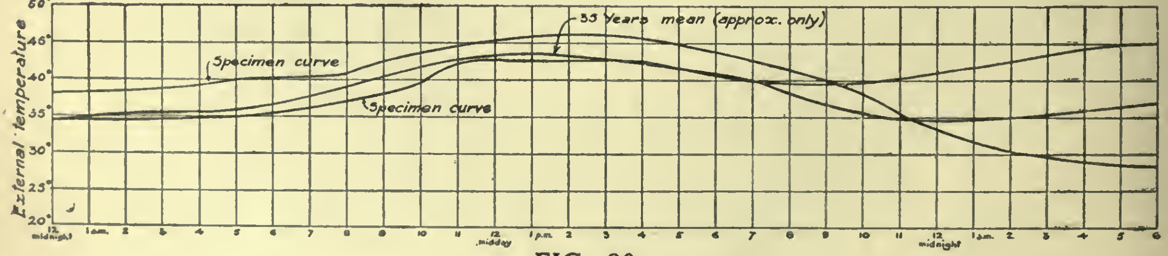


FIG. 90.

Such accurate and continuous observations can only rarely be obtained, and if obtained would need the most complicated mathematical analysis before any definite results of a scientific character can be deduced from them if, indeed, such deductions can be made at all.

Assume, for instance, that throughout a day an external temperature has been observed, varying from 35° to 45°, that heat has been continually supplied to the building during the daytime for a fortnight before the observations are taken, and that a fairly regular temperature of from 55° to 60° has been maintained in the room. The outside walls will have gradually assumed a temperature varying from 45° to 55° at the inner surface. The temperature in the thickness of the wall is lower towards the outside surface. During the night suppose the external temperature falls to 25°, the heat supply being, as usual, reduced during the night.

The temperature in the room itself only falls very gradually, owing to the amount of heat retained in the walls. If in the evening at six o'clock the internal temperature is 60° the reduced supply of heat commencing at eight p.m. by the boiler fire being banked up at that hour. The room temperature then gradually falls to perhaps 45° or 48°, owing to the amount of heat retained by the walls, which only slowly parts with this heat. By the morning a wind may have got up and simultaneously the external temperature may have risen to, say, 40°, the fire being roused at seven o'clock. In these circumstances the temperature in the apartment may be expected to be, say, 55° at nine a.m.

It is easy to see that in such circumstances it is impossible to define the precise effect of the heat supplied to the room, and impossible to produce from such results any reliable data relating to the conductivity of walls or other similar matters, which it is extremely desirable that the heating engineer should know. To calculate by any system of averages would be absolutely misleading. But such are the conditions under which a heating apparatus is ordinarily worked, and even reliable and accurate scientific calculations, if they could be made, would not of themselves furnish any guarantee of satisfactory working in the absence of the all important factor of experience.

It is not only the variations of the external temperature which cause the difficulty.

What may be called the regenerative effect of the heat retained in the wall of a room also masks to a large extent the immediate effect of the heat supplied to the room. However exact the observations which may be made, it is an extremely difficult problem of mathematical analysis to determine the relation in any given case between the amount of heat supplied to a room and the temperature changes effected thereby.

It is not surprising, therefore, that the most careful theoretical calculations may be in many cases considerably in error. Indeed, the amount of heat necessary to warm satisfactorily any particular room can properly only be determined by wide experience and careful calculations combined.

The heating engineer is not only faced with the necessity of calculating heat losses, but also as matters stand at present of guaranteeing results which it can easily be shown are impossible to calculate with any degree of accuracy. He must be prepared to make good his errors of judgment out of his own pocket, and to face the probability and the discredit of having apparently made mistakes, when, in fact, his only fault may have been that he has relied too closely on calculations. All calculations must be made bearing these hard facts in mind. His object is to provide just such an excess of heat as will keep him on the safe side, and no greater excess than necessary. If the factors he used are good they will provide for that necessary excess in ordinary cases. In extraordinary cases he has to supplement that excess by an additional percentage which his experience tells him will provide for the unusual rooms on the same scale as the ordinary ones.

The heating engineer who attempts to carry out important contracts without making as careful calculations as possible in the present state of knowledge, and in the present state of building construction, is as unwise as he who attempts to do the same thing without using a good deal of experienced judgment in addition to his calculations.

Without close calculations and careful observation combined it is impossible to learn by experience in any real sense. It cannot be called learning by experience to make a mistake twenty times in succession, unless one learns in each case by calculation and makes careful note of the character and magnitude of each separate mistake. The character of the judgment which the experienced engineer should bring to bear on each problem should be in the nature of a percentage of the calculated results. His experience should enable him to detect the danger points and the safe points in schemes for which he is responsible, and provide for them judiciously.

It must not be forgotten that his chief business is to obtain his order at a price which will enable him to satisfy his client at the smallest possible expense to himself. A client is not always satisfied when the guaranteed conditions have been technically fulfilled.

It is only in so far as the problem can be treated as one of practical science that it can be usefully dealt with in the present volume.

In this country the conditions generally specified are an external temperature of 30° in still air, with an internal temperature of 60°. In America the external temperature is generally taken as zero F., and the internal temperature at 70°. On the Continent, generally 10° external temperature and the internal temperature of 68° F.

The Effect of the Temperature of the Walls of a Room.

A room cannot be said to be heated satisfactorily until the walls as well as the furniture and other contents of the room are all raised to a temperature within a few degrees of the air temperature. If the air is at a much higher temperature than the walls, a feeling of stuffiness or parchedness is produced and at the same time a feeling of cold, especially near the floor. A large difference of temperature is produced between the air near ceiling and the floor.

Another effect of the walls being at a much lower temperature than the air is to cause down draughts against the wall. These draughts are actually rather cold, and feel colder than they really are, on account of their fairly rapid movement. On reaching the floor they are diverted horizontally along the floor and cause the feet of the occupants to feel cold.

All moving air feels colder than it really is on account of the fact that it abstracts heat at an increased rate from the body on which it impinges, especially from the body of a human being, the surface of which always carries a certain amount of natural moisture. The evaporation of this moisture with a consequent absorption of heat is accelerated by any current of air.

For these causes a room of which the air is at 60°, the walls being at 40°, will feel colder than if the air and the walls are, for instance, both in the neighbourhood of 50°.

The result is doubtless also due to some extent to alteration in the interchange of heat normally radiated in a well warmed room between the human body and the surrounding walls.

If there is a large difference between the temperature of the air and that of the walls, the balance of the interchange of radiation is, of course, in favour of abstracting heat from the body at a more rapid rate than the body is accustomed to feel in a properly warmed room, and this no doubt intensifies the feeling of cold and discomfort caused by cold walls.

Time Lag in the Heating of Buildings.

The absorption of heat by a wall is only a comparatively slow process. If it were possible to obtain a prolonged period of constant outside temperature—so long that the walls of a building should attain exactly the same temperature as the outside—and if, these conditions being secured, a constant supply of heat were delivered into the interior of the building exactly equal to that required to provide for the heat losses at the desired temperature, the actual effect on the walls of the building would be very slow.

It would normally occupy several days before the interior of the building reached the temperature at which the walls would transmit to the outer air the same quantity of heat as was being delivered in the interior after the heat lost by air interchanges has been allowed for. This length of time would increase greatly with the thickness of the walls.

When a heating apparatus is first started in any cold room, the first effect is to set up a circulation of the air from the radiators and gradually to raise the temperature of *the air*. The heat from the air is gradually communicated to the walls and the furniture. It is always many hours, and in the case of thick walls, many days after the turning on of the heat before the walls have obtained the temperatures which they will ultimately attain, or a temperature sufficiently high to maintain comfortable conditions in the rooms.

Assume that the room in which the experiment is being made is exposed on all sides to the outer air. Assume the size of the room to be 24′ × 25′ × 14′ high, of 8,400 cubic feet contents, having 60 square feet of glass surface, 1,312 square feet of outside wall 14″ thick, 600 square feet of ceiling, and 600 square feet of floor.

If the walls are assumed perfectly airtight, so that there is no interchange of air, the amount of heat lost per hour would be roughly 16,500 B.T.U. per hour when all is warm. The size of radiator surface necessary to maintain a 32° to 60° result at a mean temperature of 160° would be about, say, 117 square feet when calculated according to the usual co-efficients which are, in fact, considerably in excess of the theoretical values.

When the heating is started there are altogether 1,912 square feet of cold surface, excluding the floor, at a temperature of 32°, and 117 square feet of hot surface within the room at a mean temperature of 160°.

Even assuming that the cooling effect on the air of the surface of the walls and ceiling follows the same law as, and that the co-efficient is no greater than, that giving the heating effect on the air of the surface of the radiator, and that the walls are absolutely airtight, this would mean that the temperature of the air, shortly after the commencement of the heating, may be found from the following equation, where x is the temperature of the interior air, a the surface of the radiator, and b the interior surface of the walls.

$$a \times (160 - x) = b\,(x - 32)$$

from which

$$x = \frac{160\,a + 32\,b}{a + b} = \frac{160 \times 117 + 32 \times 1912}{2029} = 39^\circ$$

The difference $(x - 32) = 7^\circ$ (which is considerably in excess of what would be obtained in practice) is the temperature head which is forcing heat into the cold walls at the commencement of the heating. Under the influence of such a small head the temperature of the walls will rise only slowly. The air temperature cannot rise to 60° until the interior surface of the wall has a temperature which on the same assumptions as before may be calculated from the equation

$$a\ (160 - 60) = b\ (60 - x)$$

or

$$117 \times 100 = 1912\ (60 - x)$$

or

$$x = \frac{1912 \times 60 - 11700}{1912} = \frac{114700 - 11700}{1912} = \frac{103000}{1912} = 54^\circ$$

neglecting for simplicity in so approximate a calculation the difference between the inside surface temperature of the wall and window.

Calculating from these figures the amount of heat per hour which will find its way at first into every square foot of wall will be about 11 B.T.U. per hour.

If these walls are 14″ thick, the weight of this room alone will be approximately 210,000 pounds, having regard only to the walls and ceiling.

Now when the building is properly warmed and the air has attained the desired temperature, say 60° inside, the interior surface of the wall will thus be, say, 6 degrees below the temperature of the inner air, and the outer surface will be, say, 6 degrees above that of the outer air. The mean temperature of the wall will therefore be 45°.

If the building is constructed of brickwork, the amount of heat necessary to raise the walls up to this temperature will be 820,000 B.T.U. It is *impossible*, therefore, that the temperature shall be maintained in this building with the assumed radiator until the walls have this temperature, for if the walls are at any lower temperature than this, they would exert such a cooling effect on the air that it would be impossible to raise its temperature sufficiently. Even if the whole of the heat given off from the radiators were retained in the walls, and if the walls lost no heat during the heating process it would take 40 to 50 hours to raise the temperature of the walls alone.

If the *air* is to be raised to 60° within a comparatively short time the amount of radiation R at 160 which would be necessary would be found from the equation

$$R\ (160 - 60) = 1912\ (60 - 32)$$

or

$$R = \frac{1912 \times 28}{100} = 535$$

or about 4½ times as much as is necessary to maintain the temperature.

The assumed conditions are, of course, in practice, impossible of attainment. There is no building perfectly airtight. If air had been passing into and out of this building during the test the rise of temperature of the walls would have been still slower. The outer surface of the walls also would be losing heat all the time.

Fig. 91 illustrates the rate of rise in internal temperature in a building having thick walls and which is assumed to have been surrounded for a long period by an outer temperature of 30° F., so that all the walls and contents have attained that temperature. A constant and regular supply of heat is supplied to the building during the day and night, and the outside temperature remains uniformly at 30°. The heat so supplied is assumed to be exactly the quantity which would in fact be required to *maintain* a temperature of 60°. In these imaginary conditions the internal temperature would rise in the manner shown in dotted curve on the diagram.

If, however, instead of the heat supply being uniform during the night as well as the day, it had been, as usual in practice, interrupted during the night, the rate of rise would be as shown in the strong wavy curve. It will be easily seen that at no time during the period would the actual temperature have given any indication as to the sufficiency of the apparatus to maintain the desired temperature, and that if the apparatus were only just sufficient to maintain 60° under the conditions generally assumed in calculation, the temperature of 60° would never be attained at all if the heating were interrupted.

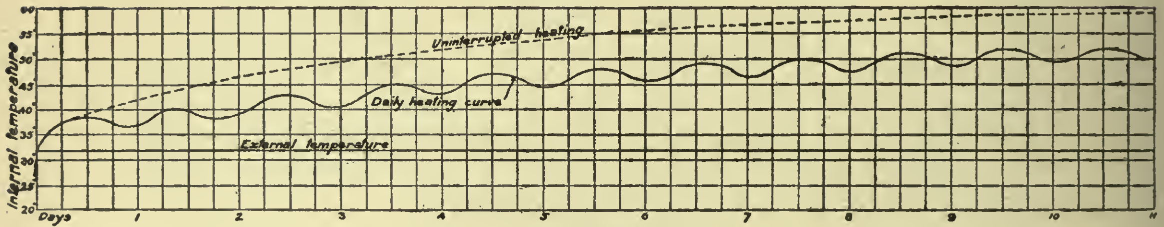


FIG. 91.

It is only on very few occasions in the course of a decade when the outside temperature remains sufficiently constant for any reliable results to be obtained by observations of this character, and no experiments have, so far as the writer is aware, been made on these lines. The actual conditions complicate the matter very much indeed, as explained in Chapter XVII.

There is in general a periodic rise and fall of temperature throughout the day and night.

Fig. 92 shows what may be taken as representative temperature charts for periods of three whole days of 24 hours in winter.

If a building were left exposed and unheated for a sufficient time after a succession of such days, it may be assumed that the temperature of the walls will be approximately the mean temperature of the air surrounding them during the previous fortnight.

If now a constant supply of heat is introduced into the building, the amount of variation in the internal temperature which will be caused by variations in the external temperature is almost impossible to calculate without the aid of very complicated mathematics indeed.

The walls have a kind of regenerating or steadying influence on the internal temperature. They only slowly absorb heat and slowly part with it.

It is essentially true that the walls *cannot* be raised up to the proper temperature without the expenditure of an amount of heat corresponding to their heat capacity. This is a factor which is entirely lost sight of in orthodox calculations, and of which no account whatever has hitherto been taken in current orthodox practice.

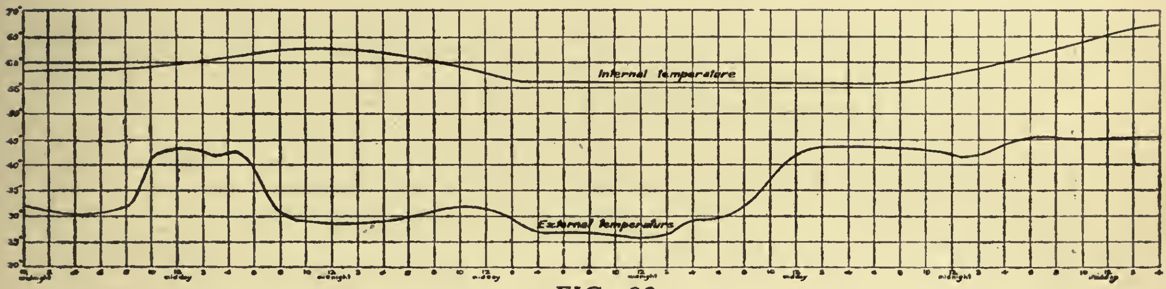


FIG. 92.

The total quantity of heat, therefore, that must be delivered into the room in order to get it into a satisfactory condition as regards temperature must have been sufficient not only to raise the temperature of the air which has been passed into and out of the room by the natural interchange, but also that of the walls, and to provide for the gradually increasing loss of heat from the outer surface of the walls while they are being raised in temperature.

In order, therefore, that satisfactory conditions of temperature may be attained within a reasonable time, it is necessary that the actual heating power shall be considerably in excess of the power that is necessary to *maintain* the temperature once it has been *attained*.

The amount of this excess may be calculated as follows :—

Let W be the weight of the walls which it is necessary to raise in temperature in order that the room may be heated.

The average temperature of the walls when the building has been satisfactorily heated may be taken to be

$$\frac{t_i + t_o}{2}$$

Let H be the excess heat required per hour necessary to heat the walls only, assuming it is all retained in the walls.

Let S be the specific heat of the material of which the walls are made, then the quantity of heat

$$= \frac{W \times \left(\frac{t_1 + t_0}{2} - t_0\right) \times S}{T} = H$$

where T is the number of hours within which it is desired to raise the temperature from cold. This, of course, assumes an extreme case.

The time between the attainment of the desired temperature difference between the inside and outside would be roughly equal to the time required to heat the room according to the above calculations.

It is quite evident that at no period of this test would the internal temperature give any reliable indication of the power of the apparatus to fulfil any temperature condition. Such an indication could only be obtained by comparing the internal with the external temperature throughout a long period.

There is no doubt, therefore, that a heating apparatus which is only just up to the theoretically required power would not be sufficiently powerful to attain within a reasonable time, nor to maintain at all times a satisfactory temperature, such as might not unreasonably be expected by a person paying for the apparatus.

Theoretical Calculations of Heat Losses.

The theoretical calculation of the heat loss consists in determining the superficial area of the glass, walls, roof, and floor of the room in question, multiplying each by the corresponding value of K and by the differences of temperature which are assumed to exist on the two sides of the heat losing surface. The sum of the various quantities of heat thus calculated gives therefore, not in reality the actual loss of heat which would take place from the room by conduction only under the assumed conditions, but as nearly as it is possible to calculate it the amount of heat which is found by experience to give satisfactory results.

To this is generally added the quantity of heat necessary to raise the assumed volume of ventilating air from the external to the internal temperature. The sum of these gives the quantity of heat which must be provided for the room. To the sum so obtained, a variable percentage is added by the practical man, according to the situation and orientation of the building.

This percentage is based on the fact that the engineer has found by experience that rooms of a certain exposed situation, and having a certain unfavourable orientation require more heat to warm them satisfactorily than do others not so situated. The necessity for the use of such co-efficients and percentages in itself disposes of any claim to accuracy which may be made for the values of the co-efficients K.

In making the calculation, the following observations should be borne in mind:—

It is customary to measure the walls in plan along their middle line and to multiply this by the mean height of the wall.

Windows are generally measured from the space between the brickwork, thus woodwork is reckoned as glass area. This allows a little margin for leakage of air.

The bounding surface of a room generally consists of window and walls of different thicknesses, floors and ceilings, or roofs, of different constructions, some of which are exposed to the outer air and some having rooms on the other side of them which are either heated or unheated. If these rooms are supposed to be heated to the same degree as the room in question, no account is taken of such surfaces, it being assumed that no heat will pass the wall, except in the case of ceilings, in which case the higher temperature at the ceiling causes an upward conduction of heat through the floor of the room above.

If, on the other hand, the rooms are unheated, an estimate is made of the probable temperature of the rooms. It is customary to take the temperature of such unheated rooms as half way between the external and the internal temperature, thus, with 30° outside and 60° inside a heated room, the unheated room adjoining the heated room would be calculated at a temperature of 45°.

A roof space whereof the outer surface is exposed to the outer air, the floor of the space forming the ceiling of the heated room below, is also generally taken as half way between the inner and the outer temperature.

In order that these somewhat extensive calculations may be effected in an accurate and rapid manner, they should be made on a special form, ruled as shown on the left side of page 295, which provides spaces for all measurements and calculations.

There should be no difficulty in applying the given co-efficients to any case which may arise.

Approximate Calculation.

In view of the essential uncertainty of the amount of heat lost from a room, owing to the various causes detailed above, it is generally sufficient to make a rougher calculation than this, especially where calculations are being made for tendering purposes.

A method which is found to be quite satisfactory in practice is to obtain for the room what the Author has called the "exposure factor" of the room.

This factor is the amount of heat lost from a room to the outside per degree difference of temperature. Approximately it represents the area of plain single glass, which would lose by conduction only the same quantity of heat as does the actual room when maintained at the same temperature.

In making this approximate calculation it is generally sufficient in the absence of any special information regarding the conductivity of the walls to reckon the wall as losing one quarter of the amount of heat that an equal area of glass would lose. In the same way, the ceiling is calculated as losing, if exposed to air, one-third of the heat lost by an equal area of glass. The floor being reckoned as losing one-tenth of the same.

The formula therefore is :—

$$\text{Exposure factor} = \text{equivalent glass area} = \text{window space} + \frac{\text{wall area}}{4} + \frac{\text{roof area}}{3} + \frac{\text{floor area}}{10}$$

When the roof is ceiled, so that there is an unheated space above the room, the area of the ceiling is divided by 6.

The formula then becomes :—

$$\text{Equivalent glass or exposure factor} = \text{actual glass area} + \frac{\text{wall area}}{4} + \frac{\text{ceiling area}}{6} + \frac{\text{floor area}}{10}$$

Unheated walls, if not exposed, are divided by 8 and added to the other sum.

Since the amount of heat lost from 1 square foot of glass is approximately 1 B.T.U. per square foot per degree difference of temperature, this "exposure factor," calculated in this way, when multiplied by the difference of temperature between the room and the outside air will give very approximately the total heat lost by conduction only. To this is added a figure equal to the contents of the room, which in thermal units will be sufficient to provide the heat required for two interchanges of air per hour. If the situation is such that three interchanges of air are probable, the cubic contents should be multiplied by 1.5 before the addition.

It is probable, in fact, that this formula is as closely accurate as it is possible for a formula to be in practice, in view of the large possible variations of air interchange and other matters, which cannot be known definitely. The practical calculations on this principle are conveniently made on a form of which specimen ruling is shown on the right of page 295.

The exposure factor × 30 + the contents of room = total heat lost from the room in one hour with two interchanges of air, with an outside temperature of 32° and an inside one of 60°.

The following additions should be made to this :—

For rooms with a northerly or easterly exposure, add 10%.

For rooms which are heated during the day only, add 15%.

For rooms which are heated during the day only and with long intervals between the successive periods of heating, add 35%.

CHAPTER XV.

ON THE METHOD OF DELIVERING HEAT TO ROOMS.

THE method of calculation explained in the preceding chapters probably gives with as much accuracy as the nature of the case permits, the amount of heat which is generally required in average conditions in order to maintain the temperature of a room of average construction and of the given dimensions, whose situation and ventilation are accurately known at any desired value. It now remains to discuss the methods by which this required quantity of heat can be supplied to the room.

In comparing the methods adopted, there are many other things to take into account besides the mere supply of the requisite number of B.T.U. per hour.

The ideal to be aimed at in all heating is that the heat shall be supplied at the lowest possible cost, by such means and in such a form that if it is not an active source of pleasure to the occupier of the room it shall at least be free from any positively offensive qualities.

The heat is in practically all cases derived from the combustion of fuel. Isolated exceptions to this are found in cases where it is derived from power and transmitted electrically, and even this power is generally ultimately derived from the combustion of fuel, except in cases where it is obtained from natural sources, such as waterfalls, etc.

There are in reality only two absolute distinct methods of conveying heat to any particular place, namely :—

(1) The direct, in which the fuel is conveyed to, and the heat generated in the place where the heat is required, and its distribution is arranged for by radiation, conduction, and convection to the air and walls of the apartment.

(2) The indirect, in which the heat developed by the combustion of the fuel is communicated to a carrying medium, which is circulated to the room to be heated, parting there with its heat by conduction or convection.

Even this distinction is not a fundamental one.

I.—Direct Method.

This method is carried out in many different ways, according as the means employed for the distribution of the heat within the apartment is mainly radiation, conduction, or convection.

In *the ordinary open fire* the means employed is chiefly radiation, conduction and convection also coming in to a slight extent.

In the *closed stove,* on the other hand, the means relied on are chiefly conduction and convection, radiation being only slightly employed to a certain extent as an accidental accompaniment to the process of conduction and convection.

The use of an open fire has obtained a firm hold in this country on account of the pleasant character of the heat which it gives, and of its simplicity. It is, however, very extravagant in fuel. There are few open fires which utilise as much as 10% of the heat in the fuel which they consume, the rest being thrown away up the chimney.

The open fire acts as a very efficient ventilating apparatus, since it maintains a powerful draught of air up the chimney, this air being drawn from the room in which the fire is burning.

In the closed stove the fire is entirely enclosed and generally is more or less out of sight. While the open fire is probably the pleasantest of all methods of heating, the highly heated stove is probably generally regarded as the most objectionable.

It is, however, very cheap in first cost, and is not very expensive in upkeep. A well designed stove utilises from 35% to 40% of the heat in the fuel.

The open fire and the closed stove illustrate extreme examples of the two mechanisms in the direct method of heating, whereby heat is communicated directly from burning fuel to the interior of a room, viz., radiation and conduction by contact of air.

In the case of the open fire the heat so communicated is almost entirely radiated, while with the closed stove a large proportion is conducted and convected heat.

In all cases where air is heated by contact with the hot surface, the character of the air becomes altered in some subtle way, which it is very difficult accurately to define, but very easy for persons of a sensitive disposition to appreciate by the senses.

Many persons can detect a marked difference in comfort between a room warmed by an open fire and the same room warmed to an equal degree by a radiator or any other hot surface. In the one case the air feels crisp and bracing, in the other heavy and stuffy, producing headaches and a languorous close feeling, as though the air had been in some way devitalised, although in both cases the air may be absolutely pure. The actual cause of this effect does not appear to be known.

It is possible that the difference may be due partly to the fact that the open fire produces much more efficient ventilation than does the radiator. It may also be due to the fact that the fire warms the walls of the room without warming the air, and that the actual difference in the feeling of the two rooms is due to the different proportions between the heat received by the body by radiation and conduction, and perhaps also the corresponding difference in humidity of the air actually drawn into the lungs and in contact with the body.

To the present writer the speculation appears possible that the fire warms the furniture to different degrees depending on its position and resulting in the production of stratified air currents at different temperatures. The varying reaction of these on the skin produces a feeling of freshness in the case of the open fire, as contrasted with a stove which heats the air all to the same temperature and produces a feeling of dull uniformity. Another theory of the present writer is that the effect may be due to some action of a hot surface on micro-organisms which fresh air may contain. Against this view is the fact that effects due to bacteria usually are not immediately discernible as is the effect of heated air. The entire question is in urgent need of investigation by competent authorities on hygiene.

The feeling of devitalisation reaches its extreme limit in the case of rooms heated on the Plenum system, whereby the air is warmed to a higher temperature than the room temperature, and the temperature is maintained in the room by the inflow of warm air.

A human body in a room surrounded on all sides by warm walls receives heat from those walls by radiation, possibly reflected radiation, and partly direct radiation. It loses heat by radiation and by contact with the cooler air. It may be the relative proportion between the amounts of heat communicated to or withdrawn from the body by these two agencies that affects the feeling of freshness.

It is possible also that the effect may be produced by the existence of currents of air at varying temperatures within the room and their direction. If the body of the air in the room is normally warmer than the walls, it is clear that the general tendency of the air currents will be opposite to that when the reverse condition is maintained.

A desk fan working in a room will sometimes appear to dispel to a very large extent the feeling of stuffiness which is produced by radiator heat. This goes to support the view that the feeling of freshness is probably inseparable from the movement of the air.

If the movement of the air is too rapid, it is felt as a draught and is regarded as uncomfortable. If it is too stagnant it becomes stuffy. Whatever the causes may be, it is certain that the feeling of discomfort is intensified by an increase in the temperature of the heating surface.

It is almost certain that the smell which is given off by hot heating surfaces is due to the partial charring or drying of particles of dust, micro-organisms, or impurities which exist to a greater or less degree in all air. Particles of dust probably carry about with them minute quantities of organic impurities or moisture, which become distilled by contact with the hot surfaces.

It is a well known fact that the olfactory organs have the power of distinguishing incredibly minute quantities of certain gaseous substances, and it is equally certain that the higher the temperature of the heating surface the more pronounced is the smell.

Dust.

It is almost certain that this smell is to some extent, at any rate, due to the action of the hot surfaces on the organic particles in air.

It is possible that the feeling of closeness may also be due to the same cause. When the walls are cold and the air of the room is warm, the cold walls cause a slow moving of the air downwards on the outside of the room. This down draught results in the forcing upwards of air from the floor level into the breathing plane, and it may be that the presence of this dust in the breathing air is the cause of the feeling of stuffiness.

In cases where walls are warmer than the air, the walls cause a corresponding *upward* current of air, which has the tendency to draw *downwards* the air in the breathing plane, and possibly this reduces the proportion of the ground dust, which is floating about in the air at the breathing level.

The most important difference between the operation of an open fire and that of a closed stove is possibly due to this fact. The radiant heat from a glowing fire passes through the air without heating it, except to a very slight extent, and impinges on the walls where it is either reflected or absorbed by the walls. The reflected heat again impinges on another wall, and ultimately the whole of the radiant heat from the fire is absorbed by the walls, or furniture, or bodies of the occupants of the room, and the heat is communicated to the air by conduction from these warmer bodies at a low temperature.

With a stove or hot surface, on the other hand, the amount of radiation is relatively small. (See chapter on "Conduction of Heat.") The major portion of the heat is carried away by contact of air, and the air thereby becomes warmer than the walls.

However, the cause of the entire phenomenon is distinctly obscure, and needs investigation from the medical side. Of the fact itself there is no doubt, and it is equally certain that the heating engineer must take the fact into serious consideration, with a view to overcoming all causes of complaints, to which heating apparatus of whatever kind seem to give rise.

The only definite fact which emerges, so far as it concerns the heating engineer in the present stage of science, is that the lower the temperature of the heating surface the pleasanter is the heat produced, except where the heat is derived mainly or entirely from radiation from glowing coals, in which case the higher the temperature of the heating surface the better.

II.—Indirect Method.

The second general method of communicating heat to an apartment —the indirect—consists in generating the heat at a central point in a furnace and communicating this heat by conduction through plates bounding the furnace to some fluid carrying medium which is circulated to the portion of the building where heat is required.

The medium is circulated at a temperature higher than the desired temperature of the apartment, and the heat is there transferred from the medium to the air and the walls of the apartment by conduction, radiation, and convection, generally through metal surfaces arranged in the form of tubes, cylindrical or otherwise. The medium is then, in ordinary circumstances, returned to the furnace, where it is again heated and again circulated to the apartment. There is therefore a continual transference of heat from the furnace to the apartment, the medium being used as the carrier of heat.

In some cases, however, the medium after it has had some or all of its available heat abstracted from it, is thrown away, this being generally a very extravagant method of working.

In some cases also the medium employed for transference of the heat is air itself, and in most cases the heated air is used to displace the cold air of the apartment, or the cold air of the apartment itself is circulated to the furnace. This is the point of contact, or halfway house, between the direct and the indirect systems.

The sub-divisions of system two consist merely in the employment of different media, and of different types of heating surface, and of different methods of causing its circulation. In very rare cases is any other medium than air, water, or steam employed.

The distribution of the heat after its communication to the air is considered in the chapter on "Air Currents within a Room." The general theoretical principles involved in the conduction of the heat are dealt with in the chapter on "Conduction."

In this chapter it is proposed to deal chiefly with the practical aspects of the matter, and especially with the comparison of media and of radiating surface (improperly so called).

Relative Advantages and Disadvantages of Water and Steam as Heating Media.

Temperatures of the Medium.

Broadly speaking, the most important practical differences between water and steam as heating media is that water is generally at a lower temperature than steam, except in what is known as the high pressure system of water heating, in which cases the temperatures of the pipes may be equal to that of steam at the highest pressures.

The advantage of being able to raise a heating surface to a high temperature is that a much smaller amount of heating surface will suffice to deliver a given number of thermal units per hour than if the temperature is lower. This leads to economy in first cost. This advantage is particularly noticeable in climates where the winter temperature is sometimes very low. In such cases the mere first cost of water heating surface to provide for the maximum requirements would be so high as to be in many cases prohibitive.

Although the first cost of steam heated surface is generally lower than that of water heated surface for the same duty, the cost of upkeep is necessarily higher for two reasons :—

Economy and Upkeep.

(1) There is general, though not universal, agreement that the efficiency of a steam boiler is lower than that of the same boiler used as a water boiler, working of course at a lower temperature, and therefore a greater number of heat units are usefully employed per pound of fuel used.

(2) Steam heated surface must always be maintained at a high temperature, and therefore for days when the outside temperature is not very cold the amount of heat delivered by a steam heated apparatus, which is adequate to supply the heat required on a very cold day, is almost of necessity greater than is required on a milder day.

Therefore a waste of fuel is produced over and above what is necessary to obtain the desired effect.

The method of keeping the room temperature down to the desired value in this case is either to shut off the valve at intervals or to waste the superfluous heat by opening the windows and increasing the ventilation more than would otherwise be done. Though this is by no means a disadvantage from a hygienic point of view, it cannot be recommended as an economical proceeding.

With water heated surface, on the other hand, the temperature of the whole of the radiators can be controlled by the simple process of reducing the combustion of fuel in the boiler, whereby the temperature of the water circulating in the entire installation is reduced. If the pipe work is correctly designed the temperature will be more or less uniform throughout the entire system at all temperatures. It will be shown in a later chapter that by merely controlling the strength of the fire on a mild day the heat supplied to all points in the entire building can be controlled and regulated in proportion to requirements.

This is not the case with a steam heated system. If it is attempted to control the heat supplied by reducing the combustion of fuel in a steam heated system, the result will be that certain of the radiators, particularly those favourably placed with regard to the boiler, will take all the heat generated, leaving the more unfavourably situated radiators quite cold.

Some radiators in the system may be half heated, that is to say, a certain proportion of the tubes in the radiator will become hot, the others being luke warm, or nearly cold. Other radiators, however, will be quite cold, and there is no sort of uniformity possible throughout the entire system.

It is not possible so to design a system of steam pipes that there will be a uniform distribution at all pressures, the distribution of the steam at low powers appearing to be largely a matter of chance, and being in any case very capricious and uncertain.

A further disadvantage of steam radiation is that it gives off a distinct smell, which has been already alluded to. This smell can be detected from radiation at as low a temperature as at 140° F. It is not very noticeable below this temperature, although certain persons can perceive it at any temperature, however low. At any rate, there can be no doubt that, especially for a mild climate, water heated surface is superior to any form of steam heated surface, from the point of view of comfort.

Frost.

The pipes in a water installation are, of course, completely filled with water. If, therefore, through the action of frost, a plug of ice becomes formed in two places in the pipes of a water installation, and subsequently the water between these two plugs freezes, the effect must inevitably be that the pipe, howsoever strong, must be burst. The two end plugs of ice, if of any considerable length, adhere so tightly to the walls of the pipes that it requires a greater pressure per square inch to force them along the pipe than it does to burst the pipe itself.

The freezing of water is, of course, attended by expansion, that is to say, the volume of the ice formed from a given mass of water is greater in bulk than the water itself, and water when subject to great cold is capable of exerting almost indefinitely great pressure.

Nothing, therefore, practically speaking, can save a hot water apparatus from serious damage if frost is allowed to get at it.

The steam system is not generally nearly so liable to damage from frost as is a water system, because steam pipes are normally, partly at any rate, filled with air. The freezing of any water there may be in them therefore does not necessarily rupture the pipe, but merely compresses to a slight extent the air within the pipes.

The case is of course different with the return pipe, when this is on what is known as the "wet return" system. These pipes are in a somewhat similar condition to the pipes of a water installation, and just as much care must be taken that frost does not reach them as in the case of a water pipe. Of course, so long as the heating apparatus is actually working there is normally little danger of frost, for the pipe is continually receiving a supply of hot condensed water, which in most cases keeps its temperature above freezing point, though not always.

Cases are by no means unknown of a long condensed pipe on the wet return principle becoming frozen solid while the apparatus is at work, and such pipes therefore should be carefully protected from frost. But in all cases a water installation is more liable to danger from frost than is a steam installation.

Noise.

A steam installation has further the disadvantage that it is liable, unless very carefully fixed, to give rise to loud cracking noises in the pipes, especially when steam is first raised in the boiler. As steam is raised generally at an early hour when the inhabitants of a private house are asleep, this is a very grave disadvantage.

These noises can however be reduced or entirely suppressed by the careful design of the pipes. They are due to the simultaneous presence in the pipe of more or less cold condensed water and moving steam, the result being probably brought about by the globules of steam becoming surrounded by cold water and a consequent "implosion" of the globules, due to sudden condensation.

There is, however, some uncertainty as to the actual cause by which these loud noises are produced. Some authorities believe them to be due to the hurling of water in the pipe by the rushing steam at a high velocity against the end of the pipes or bends, or other obstructions. Whatever the cause, the shocks produced are of such violence that in severe cases they have been known to produce disastrous explosions and rupture thick pipes. It is of frequent occurrence that they cause joints to blow and a bad leakage of hot fluid which may have very expensive consequences. Such noises are, of course, entirely avoided in the case of a water apparatus properly managed.

Overflowing of Expansion Tank.

With a water apparatus, which is allowed to boil or attain too high a temperature, not only may similar noises and effects be produced, but also the generation of steam within the body of water may cause an open expansion tank to overflow, producing extensive damage.

However, with a properly proportioned water apparatus, and with anything like reasonable management, such accidents are, if not unknown, at least of very rare occurrence.

If a boiler is more powerful than it should be, or if a large amount of the radiation is shut off in a system having a boiler of normal size, the same results may occur, but they do not occur otherwise than by carelessness and mismanagement.

Rapidity of Regulation.

Another distinction between water and steam installations is that the steam installation is much more rapidly regulated, and responds much more rapidly to alterations in the rate of combustion in the boiler. The reason for this is obvious, namely, that the weight of water contained in the water pipes has a large capacity for holding heat, and therefore the temperature in a water installation both rises and falls more slowly than in a corresponding steam installation. This may, in different circumstances, be an advantage or a disadvantage.

The thermometers in a freely ventilated steam heated room can be distinctly affected within 10 minutes or a quarter of an hour, if all steam is shut off, but in a water heated room, four or five times this period is necessary.

In such a building as a greenhouse, where a constant and slowly moving temperature is an essential, a water heated surface therefore has a great advantage on this ground. But in a room which it is required to heat rapidly, this feature of a water installation is a great disadvantage.

All the above relative disadvantages of a steam installation as compared with a water one, are much greater in the case of a milder climate than of a severe one, or rather the essential disadvantages of a steam installation are more counterbalanced by the corresponding advantage of greater power and less liability to danger from frost during very severe weather, and the more severe the climate the greater is the preference generally accorded to steam as against water.

In the British Islands water installations are in an overwhelming majority, low pressure steam only being adopted in rare instances, and that generally on the ground of economy and first cost.

We shall next consider the advantages and disadvantages of different types of radiating surface.

In recent times it is customary to expose the radiators within the rooms to be heated, entirely without protection or concealment. The radiators are made of artistic shape so that they do not constitute an eyesore. Up to within a comparatively recent date radiators were made of such ugly shapes that it was necessary in high class installations to enclose them in expensive casings which concealed the radiator, although the casings were commonly considerably more unsightly than the radiators themselves.

Recent investigations, however, have proved the dangers to health that are inseparable from an accumulation of dust, such as was general within the coil cases. These have therefore been generally abandoned and the radiators have been fully exposed.

Later improvements in the radiators have separated the individual sections so far that a hand and duster can easily be passed between them for the purpose of keeping them clean, and have lifted them some distance from the floor, either by long legs, or preferably by supporting them on brackets built into the wall, so as to enable the accumulations of dust underneath them to be readily removed. They can now be obtained of every conceivable height, width, length, and type to suit any situation.

The object aimed at in supplying a system of radiators should be to deliver the requisite quantity of heat at as low a radiator temperature as possible, that the radiators should be of such shape and fixed in such a position as to be, if not a positive ornament, at least not objectionable from the artistic standpoint. In some cases they serve a useful purpose in the decoration of a room, in filling up a blank space which would otherwise be ugly. The pipe work being so arranged as to be as inconspicuous as possible, and of such size and design that the requisite quantity of water per hour would be circulated to each radiator.

Heating by Hot Pipes Fixed in the Room to be Heated.

One of the cheapest and in some respects one of the most satisfactory methods of heating a room is by fixing pipes of as small diameters as possible in or around the room, the pipes, of course, taking the place of the radiators, to which allusion has previously been made. The chief objection urged against this method is its unsightliness. Many people have a strong objection to the appearance of a pipe, howsoever neatly fixed, and it is for this reason that radiators have been introduced this past 15 or 20 years to take the place of pipes.

There is no doubt that the old form of pipe, the greenhouse variety, is distinctly unsightly and inconvenient in many classes of rooms, particularly private apartments, sitting-rooms, and living-rooms generally. They are not beautiful in any position in any room.

Pipes of a small diameter may frequently be fixed in a position where they are not unsightly, where they are in fact very inconspicuous. The inhabitants of a room speedily become accustomed to the appearance of such pipes, and after the first day or two they are never noticed, indeed, they are less conspicuous than radiators, which themselves are, from an artistic point of view, not objects of beauty.

A disadvantage of pipe heating which, however, in certain cases may be an advantage, is that they contain, if of considerable diameter, a large mass of water per square foot of available heating surface. They therefore take a long time to heat up or to cool down. The regulation of temperature in a room heated by such pipes is therefore slow. It is further not easy to fix the exact quantity of pipe in a room which the circumstances demand, without, in some cases, stopping short a range of pipes in the middle of a wall, which adds to the ugly effect.

It is in some cases an advantage in that the temperature in a room provided with them cannot vary rapidly. This feature is not always attainable in a radiator heated apartment, especially a steam radiator heated apartment. If the boiler fire is allowed to become a little low the effect is felt in a quarter of an hour in a steam heated building, in a much longer time in a water heated building, because of the considerable bulk of water in the pipes.

One great advantage of pipe heating of a room is that the heat is equally distributed all over the wall instead of being concentrated in one spot. This is particularly advantageous in such buildings as schools, where the children are seated all over the room. Radiators fixed at the back of a child, or group of children, or close to them on the side, are apt to produce in delicate children coughs and colds, which a pipe system of heating avoids, owing to the heat being equally distributed all over the wall, and its not being felt in any one particular place more than another. Also in buildings of this class the appearance is not of much account, so that in the opinion of the present writer pipe heating for schools and other similar buildings is preferable to radiator heating, though architectural opinion is generally unfavourable to the use of pipes in any form, and is favourable to the concealment of such pipes as cannot be avoided.

On the other hand, the concentration of heating surface in one place in a radiator may be made to correspond more accurately with the positions in a room, *e.g.*, under windows, where there is the greatest loss of heat and therefore the greatest danger from cold draughts.

Cheapness.

The rate of heat emission from pipes heated by any medium is greater, square foot for square foot, than the heat emission from any known class of radiator. For this reason a smaller number of square feet of pipe surface will do the work of a larger number of square feet of radiator surface, although radiators can be bought so cheaply, that at the present time it is commonly cheaper to put in a large amount of radiator surface than the corresponding equivalent of pipe surface.

Table XV. and Fig. 93*a* give the heat emission from a pipe surface which is thoroughly exposed to the air. The values here given are not so liable to fluctuations as the corresponding value for the radiator surface, because of the greater uniformity of the condition in which pipes are generally installed. One can reason, for instance, that 100 feet of pipe will give out double the amount of heat that 50 feet will at the same temperature, except when pipes are fixed in close proximity one to the other, in which case reduction of efficiency may be calculated on, as shown in the table.

From the figures given it should be easy to calculate the amount of radiation at any given temperature which should be required for any given heat emission.

General Remarks on Arrangement of Radiators and Apparatus.

The heat which is delivered to the radiators by a circulation of the hot fluid is transferred to the apartment by means of convection currents of warm air. This warmed air gradually heats the wall and brings the room as a whole to the desired temperature.

Owing to the effect of a hot surface on the feeling of freshness of the air in a room, where economy and first cost are not a matter of the first importance, it is very advisable that these radiators should be kept at the lowest temperature consistent with delivering the required quantity of heat.

In this climate there are only very few days in a year in which the temperature is very low, the temperature on the great majority of days having a medium value from 40° to 50°. If, therefore, full provision is made for adequate heating with a very low outside temperature and a moderate temperature in the radiators, the installation will be much more powerful than is necessary for the great majority of days for which heating is required.

In these circumstances, however, the heat can be maintained by a very low radiator temperature, which is in itself a great advantage from the point of view of comfort, although it multiplies the expense of the installation.

In any case, in a climate such as this, it is very desirable that it should be possible to regulate the temperature of the radiators as a whole from a very high to a very low value.

For an installation of any considerable size this is best done by dividing the boiler power into two parts, having a ratio, say, 2 to 3, or 1 to 2. If this is done the adjustment of the power of the apparatus, according to the value of the outside temperature, is very simple.

Positions and Sizes of Radiators.

The radiator surface in any room of considerable size should be divided into two or more parts. The greater the number of parts the less is the difference in temperature between the ceiling and floor.

Where a large room is heated by one very powerful radiator, the strong up current of air from it causes a layer of air at a relatively high temperature to accumulate at the ceiling, and a great difference in temperature between ceiling and floor which gives rise to complaints of cold feet.

The greater the horizontal distribution of the radiators the less is this difference. A proper portion of the heating surface should be placed under windows where the greatest want of heat exists. The cold down draught from the windows and the hot upcast from the radiator neutralise one another and reduce the difference of temperature.

A feeling of comfort and freshness in a room depends largely on the movement of the air, such currents of air being at a temperature and moving with a velocity not great enough to be felt as a draught.

If there is no movement, complaint is made of stuffiness. If there is too much movement, or if the moving air has too low a temperature, complaint is made of draught.

Special care must be exercised in the placing of radiators in cases where rooms are very low.

It has been pointed out that the effect of the radiator is to give an upward current of warm air, which spreads out over the ceiling. If, therefore, a low room is provided with projecting horizontal beams under the ceiling, these will prevent the uniform distribution of the warm air over the ceiling, and will cause that part of the room between the beam and the radiator to become considerably hotter than the rest of the room.

Similar considerations apply to balconies in churches and halls. The space under such balconies is not sufficiently heated by powerful stoves or radiators placed in the middle of a hall or church, because the convection currents do not carry the cold air out and the warm air in sufficiently rapidly.

On the other hand, if the space underneath the balcony is too liberally provided with heating surface it easily becomes over heated, a circulation is set up in the space underneath the balcony which does not generally reach the body of the church.

All such spaces, therefore, should be provided with means for the escape of the heated air at the back of the balcony up into the space above the balcony, gratings and flues being provided in the walls.

A large church or hall needs particular care in order that draughts may be minimised. The upper part of such a building is not infrequently left quite cold, and any considerable body of cold air at a high level will infallibly fall and will be felt as a draught by the persons sitting underneath it.

In these cases it is almost always necessary to provide a considerable portion, say, 25% to 30% of the whole radiation at a high level.

When consideration of the design for such a hall leads to the view that a down draught will probably take place at the particular part, care should be taken, where possible, to break it by means of a horizontal ledge placed at a suitable height. If this cannot be done, heating surfaces must be provided to neutralise it.

The effect of a radiator placed under a large window is that the mounting stream from the radiator meets the descending cold stream, forming a resultant stream in a horizontal direction, which not only distributes heat, but reduces its intensity.

It is frequently advantageous, in order to prevent down draughts from a window under which a radiator is placed of insufficient power, to place a baffle plate at the top, so that the downward current from the window is induced to flow to the back of the radiator, and is warmed before it comes into the room.

The necessity for radiators being so fixed that they can be kept clean is alluded to in another place.

For this reason radiators having plain surfaces are much to be recommended, and vertical surfaces are to be preferred to horizontal ones.

The old fashioned coil cases containing horizontal coils of pipes have been almost completely abandoned in the present times, owing to the facility with which they collect dirt and dust.

In cases where coils are used for concealing purposes, it is imperative that they should be so arranged that they can be immediately and easily removed for cleaning purposes.

The use of radiators inevitably involves the distribution of the heat by currents of air, and the condition that the air in the room is warmer than the walls. This is probably to a great extent the cause of the great objection that is often taken to the use of central heating apparatus as against open fires.

CHAPTER XVI.

THE EMISSION OF HEAT FROM RADIATING SURFACE.

STEAM and water are the only two fluids in common use for conveying heat to radiating surface. Air is sometimes used for the same purpose. This application is dealt with in a separate chapter.

When low pressure steam is used as heat carrier it is of necessity at an almost uniform temperature. Provided that the supply is sufficient and that the connections are properly made so that the surface is not either "air-logged" or "water-logged," the temperature of the entire radiator is also approximately uniform. With water, on the other hand, the temperature of the fluid in contact with each part of the surface varies.

It by no means follows, however, that in practice a surface heated by steam at a constant pressure and temperature has an absolutely constant heat emission, so far as such a result can be secured by constant conditions within the surface itself, because in almost all practical cases a certain amount of air is mixed with the steam within the radiator, which has a marked effect on the heat emission. If all air could be removed from the interior of a steam heated radiator, the emission which might be calculated on from it would be increased. It would certainly be more uniform than is the case in practice.

With a water heated radiator the rate of heat emission from each part of the surface varies, of course, with the temperature of the fluid in contact with that particular part. It is not possible to name any one temperature as being that of the whole radiator. It is a matter of the greatest difficulty to calculate exactly the amount of heat actually lost from any water radiator from the temperatures alone.

It is not possible to calculate generally the temperature of any particular part of a water radiator, nor to say how much of the total heat emitted proceeds from any particular part, except in a very simple case, such as a long horizontal pipe.

The water enters at a higher and leaves at a lower temperature. Each part of a radiator in which perfect circulation is maintained *should* have a temperature somewhere between that of the inflowing and outflowing water. In practice this condition is not maintained. The temperature of what may be called the "off corner" of almost all radiators connected above and below, that is, the bottom corner at the opposite end of the radiator to the outflow, is not infrequently, indeed usually, some degrees and often many degrees below the temperature of the outflowing water.

Conditions which Influence the Total Heat Emitted by any Radiating Surface.

The determination of the total emission of heat from any kind of radiating surface can only be made by careful experiment.

It has been shown that the emission varies according to a great many conditions, of which theory cannot possibly take account.

The same heating surface will emit widely varying amounts of heat, according to the way in which it is fixed and the general conditions under which it is worked.

The following are the chief factors which determine the amount of heat which will be given off from one square foot of heating surface:—

(1) The character and shape of the surface and the distance apart of its sections, if any.

(2) The temperature of the media in contact with it and the humidity of the air.

(3) The velocity of the media relatively to the surfaces and the way in which the circulation is effected.

(4) The surroundings of the surface, including (*a*) the kind of casing, if any; (*b*) the proximity of the wall; (*c*) the temperature of the wall; (*d*) the amount of window surface in the room; (*e*) its position relatively to the radiator; (*f*) the position of the furniture in its neighbourhood; and on several other similar things.

(5) The character of the paint or enamel with which it is covered.

It is thus obvious that even very accurate experiments may be made on an actual radiator fixed in a laboratory, and that the results may be very widely different from those which are obtained from the same radiator in practice. In fact it is impossible even by the most careful calculations, and even when these are based upon exact practical experimental observations, to specify accurately what the heat emission of a radiator will be, unless the exact conditions in which it is installed are known.

These conditions cannot in the nature of the case be known with certainty, except in special cases. It follows that there must be a good deal of uncertainty in the calculations regarding the minimum quantity of radiation necessary for any given duty, even though the number of heat units desired are known with considerable accuracy, which latter, however, is rarely or never the case in practice. If, as usual, therefore, it is necessary to be on the safe side, very considerable allowances over the calculated minimum quantity of radiation must be made.

Character of the Surface.

The amount of heat emitted by any heating surface depends to a very great extent on the character and shape of the surface, and the way in which the heating and heated liquids circulate to it. In the case, for instance, of a relatively small pipe through which a constant stream of hot water is circulated in one direction, the conditions are essentially different from what they are in the case of a radiator kept warm by a slow and constant stream of hot water passing into it, and an equal stream of cooled water passing out of it.

In the first case the water in contact with any particular section of pipe is constantly renewed or scoured away by fluid having a more or less uniform and appreciable velocity.

In the other case variations in the temperature of the water produce slight differences of density in different parts of the radiator. The resulting movement of the water is not only slower, but much more irregular than is the case with a single length of pipe. It is not sufficiently rapid to renew forcibly the layer of water in actual contact with the interior surface. The temperature of that layer may be, and probably is, several degrees below that of the body of the water.

Further, in the case of the radiator, the massing together of a relatively large quantity of heating surface raises the temperature of the air in its immediate vicinity, and therefore reduces the difference of temperature between the heating and the heated fluid on which the total heat transmission depends.

The heat emitted from radiators varies according to the way in which the surface is massed together, and especially according to the facility with which cool air can reach the hot surfaces. Indeed, the heat lost from any heating surface is so greatly influenced by the velocities of the fluids, especially that of the air, that all other influences combined are almost negligeable in comparison.

Pipe Surface.

A plain single cylindrical horizontal pipe is the simplest form of radiating surface with which we have commonly, to deal. Each part of this kind of surface is in a more or less uniform condition in relation to the temperature of the exterior air. The temperature of the water within the pipe, however, varies from point to point, a feature common to practically all water heated surfaces.

There is a symmetry about a horizontal pipe which does not exist in the case of any other form of radiating surface. No part of the pipe is subject directly to the influence of neighbouring heated parts of the same pipe. It might be expected, therefore, as is actually found in practice, that a plain horizontal pipe, especially where the diameter is small, is the most efficient of all practical forms of heating surface, and that the amount of heat obtained from it is proportional to the length of pipe where the relative temperatures remain the same.

A horizontal pipe, too, is less subject than other forms of heating surface to variation in emission, owing to its situation, or other causes. For instance, if a horizontal hot water pipe passes round a room, in which there is one particularly cold or draughty corner, the total heat lost from the pipe will depend chiefly on the *average* conditions round the whole of the room through which it passes. The heat lost is not materially altered by a localised draught impinging on the pipe at one point.

If, however, a radiator is fixed in the path of a draught, the resulting increase in the total B.T.U. emitted would be proportionately very much greater, because a large part of the surface would be subject to the influence of the increased velocity of the air.

Effect of Alterations in Diameter of Single Horizontal Pipes.

The heat emission per square foot from a pipe becomes greater as its diameter decreases. The reason for this probably is—

(1) That in the neighbourhood of a large pipe the temperature of the air is raised, and therefore the pipe does not emit so much heat.

(2) The smallness of the bore conduces to an efficient scouring action, whereby the flow of water keeps the layer of fluid in actual contact with the pipe constantly renewed, so that its temperature is maintained higher than that of a larger pipe.

The same mechanical causes which increase the friction in a small pipe make it efficient as a heating surface. With a large pipe, 4 or 5″ in diameter, or greater, there is a marked falling off in the emission per square foot, doubtless due to this cause, viz., that the main stream of water tends to flow along the core or centre part of the pipe, leaving the movement of the water in contact with the metal sluggish.

This explanation is borne out by the fact that the emission from a large pipe is considerably increased if baffle plates are provided inside it, of such a form as to effect the more thorough mixing up of the water passing along it.

Fig. 93 and Table XV. give the values of the heat lost for pipes of different diameters.

Effect of Placing Several Pipes in Close Proximity to One Another.

As might be expected, the effect of placing a number of pipes near one another is to reduce the mean heat emission per square foot, owing, of course, to the fact that the temperature surrounding the pipes is thereby increased, free access of cold air to the pipes is diminished, and a considerable part of the radiant energy is suppressed.

The heat lost from the lowest pipe of a range is not affected to the same degree as that from the upper ones.

The nett radiant energy lost from the upper part of the bottom pipe is considerably diminished by the proximity of an equally hot pipe above it, which of course radiates heat back to the lowest one, but the *total* effect on the lowest pipe is less than that on the upper pipes of a range.

Differences between the Heat Lost from Horizontal and Vertical Pipes.

When the conditions under which air circulates over them are considered it would seem that there should be some difference between the heat lost from horizontal and vertical pipes.

With a horizontal pipe, when the air in contact with the exterior surface becomes heated, it immediately rises out of contact with the hot surface. Therefore the air in contact with the surface of such a pipe must be continually changing, and the same air only remains in contact for a comparatively short time.

With a vertical pipe, on the other hand, air heated by contact with the lower portions tends to cling to the vertical surface. The air surrounding the upper portion is therefore warmer than that near the lower portion, and warmer than it would be if the pipe were horizontal.

On the other hand, the velocity of the air in contact with a vertical pipe is greater than with a horizontal one. The difference in emission, if any, is therefore less than it would otherwise be.

A further difference in the practical heating effect is that where a vertical pipe passes through a room it only heats the air in one place, probably a corner of the room. About half the total length of the pipe delivers its heat overhead, which tends to dissipate the heat where it is not required. A greater useful effect is probably obtained from a horizontal pipe which is generally fixed at or near the floor level. The full effect of the heat is obtained. The rising warm air becomes well mixed with the cool air surrounding it before it has reached any considerable height from the floor. The differences of temperature between the upper and the lower parts of the room are not so great as where vertical pipes are employed.

It would appear, therefore, that calculations for horizontal and vertical pipes should be to some extent different, but to what extent experiment alone can decide.

As in all cases where opinions depend on experience or judgment (often another name for prejudice) there is much difference of opinion on this point. Some authorities advocate a 5% reduction in the calculation for heat emission when the pipe is vertical, others consider that no difference should be made. Yet other authorities consider that the total heat emitted by vertical pipes must be calculated about 5% larger than with horizontal ones.

With steam heating it is almost certain that a vertical pipe has a greater emission than a horizontal one, owing to the greater ease with which a vertical pipe clears itself of condensed water.

A horizontal steam pipe always has a layer of water covering the bottom of the pipe. This water is in the worst possible position for conducting heat to the pipe. Any heat transmitted through it must be conducted downwards in the opposite direction to that in which the convection currents generated by the varying density of the water tend to carry the heat. In these circumstances water is an extremely bad conductor of heat. It is no uncommon thing for a horizontal steam pipe to lose 25% of its power, owing to this cause, though, of course, the part of the surface which is rendered ineffective is that part which in any case is the most inefficient, viz., the bottom of the pipe.

It is usual to reckon that a horizontal steam pipe is less efficient than a vertical one, to the extent of 5% for pressures below 10 lbs. per square inch and 10% for higher pressures.

The influence of height in vertical radiating surfaces can be neglected up to a height of three feet. With greater heights than this, however, it is recommended to decrease the calculation of heat emission by various percentages, amounting to 10% in extreme cases.

Numerical Value of Heat Transmission through Pipes.

The loss of heat from pipes has been investigated by Rietschel, who gives a series of values of K for use in ordinary practice which is almost universally adopted on the Continent. The translation of these tables into English units is given in the Appendix. The dotted lines on the left side of Fig. 99 show these values plotted on a base representing difference of diameter.

The second series of curves show the values plotted on a base repre-

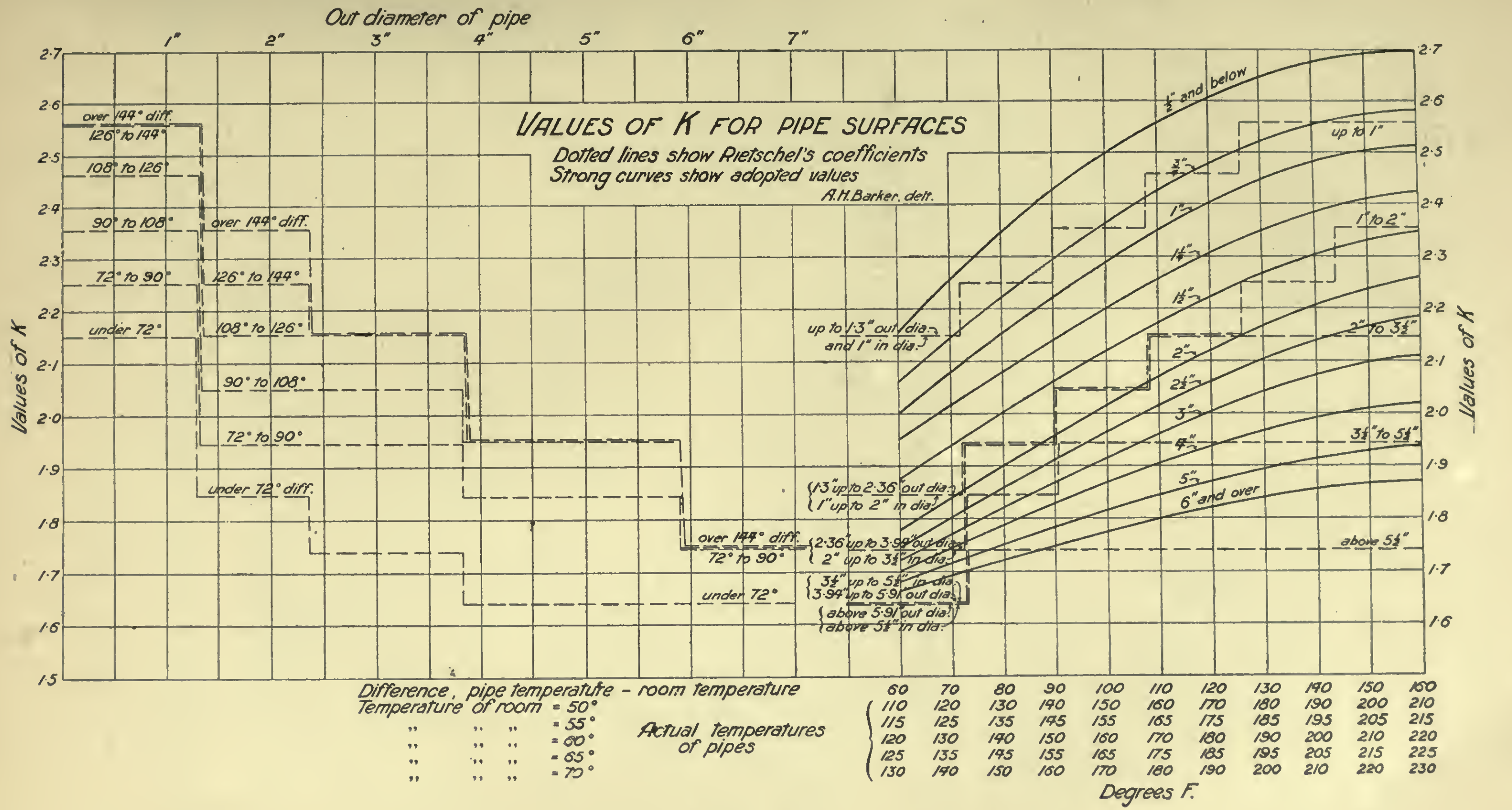

VALUES OF K FOR PIPE SURFACES
Dotted lines show Rietschel's coefficients
Strong curves show adopted values
A.H.Barker. delt.
Out diameter of pipe
1"
2"
3"
4"
5"
6"
7"
Values of K
2·7
2·6
2·5
2·4
2·3
2·2
2·1
2·0
1·9
1·8
1·7
1·6
1·5
over 144° diff.
126° to 144°
108° to 126°
90° to 108°
72° to 90°
under 72°
over 144° diff.
126° to 144°
108° to 126°
90° to 108°
72° to 90°
under 72° diff.
over 144° diff.
72° to 90°
under 72°
up to 1·3" out dia. and 1" in dia.
1·3" up to 2·36" out dia. 1" up to 2" in dia.
2·36" up to 3·99" out dia. 2" up to 3½" in dia.
3½" up to 5½" in dia. 3·94" up to 5·91" out dia.
above 5·91" out dia. above 5½" in dia.
½" and below
¾"
1"
1¼"
1½"
2"
2½"
3"
4"
5"
6" and over
up to 1"
1" to 2"
2" to 3½"
3½" to 5½"
above 5½"
Difference, pipe temperature – room temperature
60 70 80 90 100 110 120 130 140 150 160
Temperature of room = 50°
" " " = 55°
" " " = 60°
" " " = 65°
" " " = 70°
Actual temperatures of pipes
110 120 130 140 150 160 170 180 190 200 210
115 125 135 145 155 165 175 185 195 205 215
120 130 140 150 160 170 180 190 200 210 220
125 135 145 155 165 175 185 195 205 215 225
130 140 150 160 170 180 190 200 210 220 230
Degrees F.

FIG. 93.

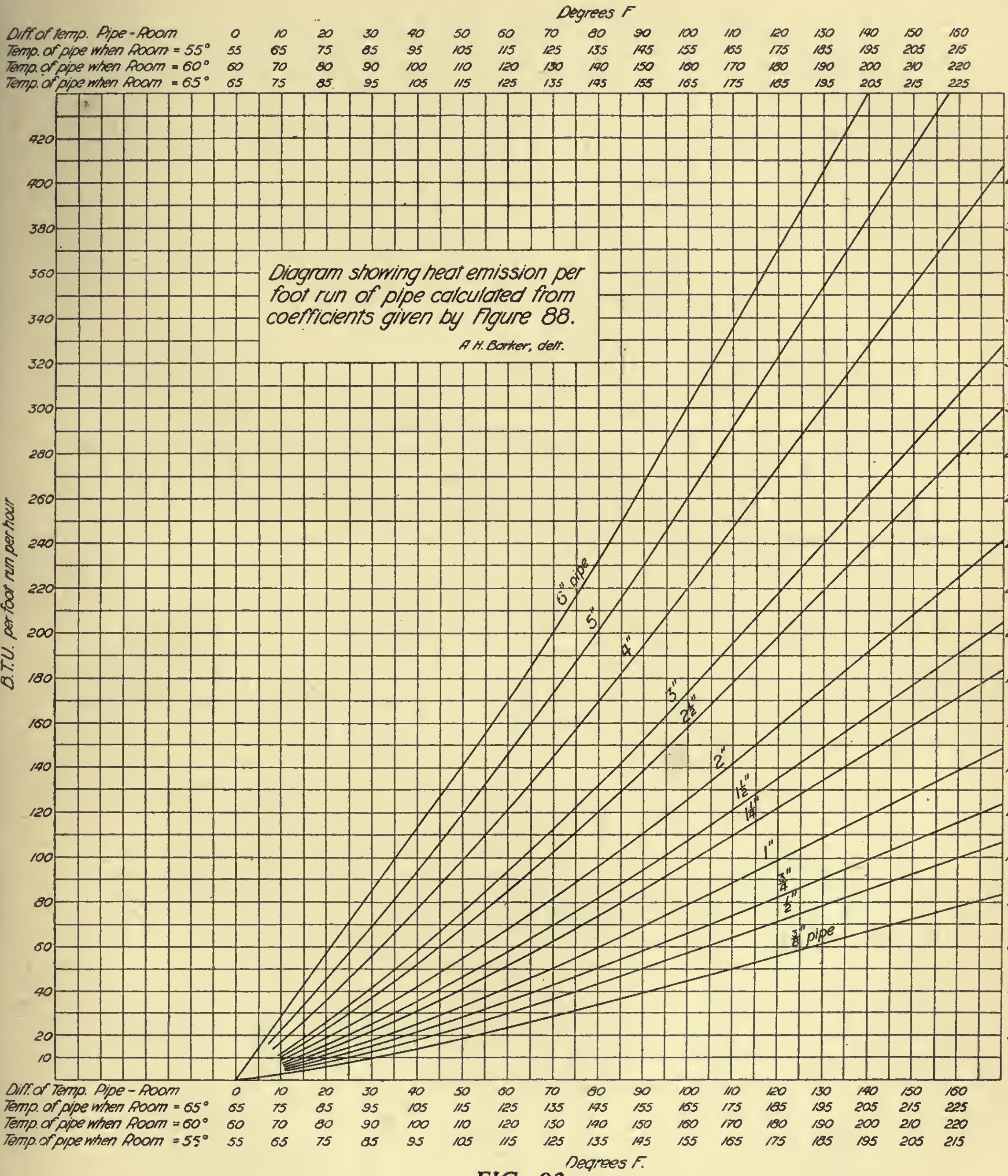


Diagram showing heat emission per foot run of pipe calculated from coefficients given by Figure 88.

A H. Barker, delt.

FIG. 93a.

senting the temperature of the pipe. It will be seen that the points lie on no regular curve, and probably no great accuracy is claimed for these co-efficients. In order that the co-efficients may be self-consistent separate curves have been assumed agreeing as nearly as possible with Rietschel's values. These are shown in the strong curves on the diagram.

It is obvious that the value of K must vary with the outside diameter of the pipe, and that it is not accurate to assume, as Rietschel does, that the value of the co-efficient can change suddenly at a particular diameter.

The given curves represent an effort to adjust these values, based on Rietschel's figures. From the values so obtained the table of total emission per foot run of pipes of various sizes has been derived.

Loss of Heat from Coated Pipes.

The amount of heat lost from the surface of a coated pipe depends to a great extent on the character of the coating employed.

Given a certain superficial area, and a certain surface temperature of the coating and the surrounding air temperature, the heat lost per square foot of coating surface is practically determined. It does not depend primarily on the temperature of the surface of the pipe, but on that of the surface of the coating.

The temperature of the surface of the pipe is only of importance in so far as it affects that of the surface of the coating.

The amount of heat lost does not depend to a very great extent on the finish of the surface, where this is of an ordinary description, so much as it does on the non-conducting qualities of the coating. Even in ordinary cases differences in the finish produce a slight variation, but not so much as to make any material difference in estimating the value of the coating.

If, however, a surface finish of a highly polished or reflecting character is given to the coating the loss of heat shows a marked radiation as explained below.

Just as in the case of all other radiating surface, the heat lost is naturally divided into two parts :—(1) That lost by conduction to the air and carried away by convection currents, and (2) that lost by radiation.

Conduction.

The amount of heat actually passing by pure conduction through the substance of the coating depends—

(1) On the difference of body temperature between the inner and outer surfaces.

(2) On the thickness of the coating.

(3) On the co-efficient of conductivity of the whole mass of material.

The temperature of the inner surface of the coating next to the pipe may be taken to be almost the same as that of the surface of the pipe, and this again will be quite close to the temperature of the medium in actual contact with the interior portion of the pipe.

The temperature which is actually maintained on the exterior of the coating automatically adjusts itself until it reaches such a point that the same quantity of heat is lost by radiation and convection together from the exterior surface of the coating, as is conducted through the substance of the coating by the existing temperature gradient.

It will thus be seen, therefore, that to increase the thickness of the coating not only increases the surface losing heat to the surrounding air, but at the same time it tends to increase the difference of temperature between the inner surface and outer surface. Although the increase of thickness does on the whole diminish the loss of heat, the reduction is not so great as it would be in the case of a flat surface. In the latter the increase in thickness of coating would decrease the temperature gradient without at the same time increasing the surface exposed to air.

In the case of a metal surface, heated by steam, the difference in total emission caused by variation in the velocity of the steam has been already alluded to (see page 257). This effect is not found to the same extent in the case of a coated pipe. The difference between the amount of steam condensed by a coated pipe when the steam is at rest and when it is in motion is inappreciable and need not be considered.

The reason for this probably is that with a smaller absolute rate of heat loss, the temperature of the material of the pipe is so close to that of the steam that small variations in this difference, such as may conceivably be produced by alteration in velocity of the steam, do not appreciably affect the total result.

Effect of the Material of the Coating.

The heat transmitted through the coating by pure conduction does not depend altogether, or even mainly, on the absolute conductivity of the solid material of which the coating is made, so much as on the way that material is massed together to form the coating. It is quite conceivable that a very efficient coating might be made by applying in a certain way a solid substance, which is itself a fairly good conductor of heat.

For instance, it is quite conceivable that three or four highly polished concentric metal cylinders surrounding the pipe and having air spaces between them might prove a very efficient non-conductor.

It has already been pointed out that the rate of transmission of heat through any surface is greatly altered by the existence of air spaces within the substance of the material. For every surface of separation between a solid and a fluid the resistance to the passage of heat is considerable.

Where the material of a coating is so applied that a large number of air spaces or cells are entrapped within the substance of the coating material, the pure conductivity of the material itself, apart from the entrapped air, affects the result only to a slight extent. The material, as a whole, comprising as it does a mixture of the solid substance with air cells, has a conductivity which differs from and is in all cases much less than the absolute conductivity of the solid substance.

It is true that each of these air cells allows the transmission of heat by minute convection currents existing within the cell itself, but these cells being of small size, the amount of heat so transmitted is a great deal less than would be transferred in the same time by conduction through solid portions of material, introduced in place of the air cells.

However, the substance of the coating as a whole, including the air cells, obeys closely the same rule as does a solid plate, that is to say, the formula explained on page 248.

$$Q = A\,H\,\frac{t_1 - t_2}{e}\,K$$

holds good, the temperatures being taken within the substance of the coating.

The value of this co-efficient of conductivity K taken throughout the material as a whole, determines the value of the coating as a heat saver, assuming the finish of the surface is the same. The character of the surface has only an effect on the amount of heat *radiated,* and very little effect on that *conducted* to the air and conveyed away by convection currents.

Heat Radiated, effect of surface finish.

All ordinary finishes of covering material, such as are commonly used in practice, give roughly the same result as regards the amount of heat radiated. Thus, for instance, a common form of cotton covering may be used, and no considerable alteration is produced in the total heat emission, if this is painted black with a polished surface, or with a dull surface, or is whitewashed, or painted, or varnished, or treated in any other ordinary manner, or even omitted altogether. The only way in which the amount of heat lost can be considerably reduced consists in applying to the surface of the coating a finish of a reflecting character.

It has been pointed out in Chapter XII. that the reflecting power of a surface is inversely proportional to its radiant power. Proof of this is found in the fact that when substances of different reflecting powers, and at the same temperature, are placed in proximity to one another within a vessel maintained at the same temperature, the mutual exchange of radiation does not produce the raising or lowering of the temperature of any of them.

If, therefore, the surface of a non-conducting composition is covered with sheet tin or tinfoil, or other surface which has a relatively high reflecting power, one would expect to find, and it is found in practice, that the amount of heat lost by radiation is very considerably reduced, so much so in fact that the entire loss of heat from the coating is diminished from 20% to 25%.

A confirmation of this observation may be obtained as follows :—

Let a pipe be covered with a uniform thickness of any quality of coating and allowed to contain steam at a slight pressure. Cover one portion of the coating with a layer of tinfoil. Insert two chemical stem thermometers into two small oblique holes made in the substance of the covering, so that the bulbs of the thermometers are as near as possible to the surface, the bulbs being entirely surrounded by the covering, and the one lying under the surface of the tinfoil and the other under the uncovered surface.

In this one it will always be found that the body temperature of the coating underneath the tinfoil is considerably higher than if there is no tinfoil. Since the inner surface of the coating must be the same in the two cases, and since the temperature under the foil is higher than where the material is uncovered, it is clear that the temperature gradient in the former case must be lower than in the latter, and since the conductivity of the substance is the same, the loss of heat is therefore definitely proved to be less, although the surface temperature of the foil is greater. If the tinfoil is afterwards covered with lamp black so as to destroy its reflecting power the difference disappears.

Relative Quantity of Heat Lost by Convection and Radiation.

The formula of Dulong and Petit is as follows :—

$$\text{Heat lost by radiation} = M\,(1.0077)^{\theta}\ \ (1077^{t} - 1).$$

$$\text{Heat lost by conduction} = N\,p^{a}\,t^{1.233}$$

where θ is the temperature of the surrounding air,

t the difference in temperature between the surface, and

p the pressure of the air, and

$a\,M\,N$ constants.

The investigation of the loss of heat by conduction through the coating is as follows :—

Assume a hot pipe of external radius r feet is covered with composition having external radius R feet. The quantity of heat passing through any cylindrical layer of the coating must be constant for reasons which have been many times explained.

Consider such a layer of radius r and thickness $d\,r$.

Let t_1 be the temperature of the surface of the pipe and t_2 that of the outside surface of the coating, and $d\,t$ the difference of temperature at the two sides of the layer.

Let W be the heat lost per square foot of the external surface of the pipe and K the co-efficient of conductivity of the substance of the material.

Then we have

$$W = \frac{K\,dt\,a}{dr} = \frac{K\,dt\,r}{r_1\,dr}$$

where a is the area of the layer at a radius r corresponding to one square foot at the outside surface of the pipe.

$$a = 1 \times \frac{r}{r_1}$$

whence transposing

$$\frac{W\,r_1}{K}\,\frac{d\,r}{r} = dt$$

Hence integrating between the limits r_1 and R

$$t_1 - t_2 = \frac{W\,r_1}{K}\log_e\frac{R}{r_1}$$

$$K = \frac{W\,r_1}{t_1 - t_2}\log_e\frac{R}{r_1} \qquad \text{I.}$$

This gives the value of K.

It is also possible by means of this equation to determine the heat lost for different thicknesses when the loss for one particular thickness is known.

The heat lost from the outside of the coating is $k\,(t_2 - t_a)$ per square foot.

But the heat lost per square foot from the outside is less than W the heat lost per square foot from the pipe in the proportion $\frac{r_1}{R}$ whence we have

$$\frac{W\,r_1}{R} = k\,(t_2 - t_a)$$

Substituting in the original equation I. and so eliminating t_2 we obtain

$$t_1 - t_a = W\left(\frac{r_1}{R\,k} + \frac{r_1}{K}\log_e\frac{R}{r_1}\right) \qquad \text{II.}$$

Curves calculated from this equation are given on Fig. 94, taken from a valuable paper on this subject by Mr. D. R. Maclachlan, read before the Leeds University Engineering Society in 1907.

Effect of Varying Thicknesses of Coating.

It will be seen that while an increase in the thickness of coating does actually reduce the heat lost from the pipe, yet it by no means increases the efficiency in proportion to the increased expense.

The question therefore arises, "what is the best thickness of coating to apply in any given case?"

The curves given on Fig. 95 show the relative amount of heat lost when the thickness of the coating is varied.

It is the common practice among manufacturers of coating to charge for coating by a rate per square foot, and this rate is commonly irrespective of the thickness, although sometimes the thickness is taken into account.

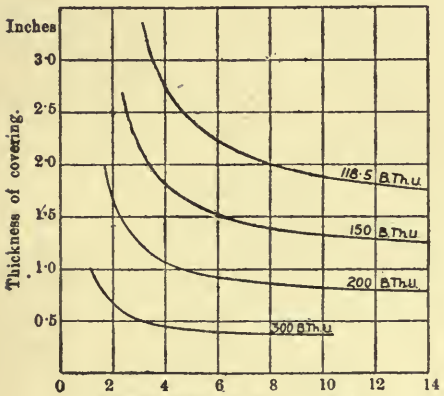


FIG. 94.

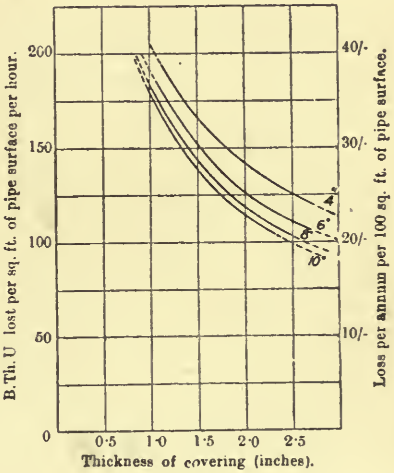


FIG. 95.

In order to compare the values of two coatings, it is necessary to calculate the saving per annum due to the employment of either for a given range of pipes, and to calculate the corresponding cost as follows:—

Assume that it is required to determine what is the most advantageous thickness of covering to coat 100 feet of 4″ pipe, comparison being made between the thicknesses 1″, 1½″, and 2″. A quotation is received as follows:—

For 1″ thick	4.85d. per sq. ft.
For 1½″ ,, ...	5.6d. ,, ,,
For 2″ ,, ...	6.1d. ,, ,,

From which it will be seen that the total cost would be as follows:—

1″ thick would be	£3	8	9
1½″ ,, ,, ...	4	11	8
2″ ,, ,, ...	5	12	6

The total condensation of steam in this pipe, if naked, would be about 43.3 pounds per hour, or 129,000 pounds per year, assuming a working year to consist of 3,000 hours. This would cost, with coal at 12s. 6d. per ton, and reckoning 8 pounds of steam to one pound of coal, in fuel alone, £4 11s. 0d. per annum.

By means of the above equation the saving to be effected by different thicknesses can be calculated and that thickness which would give the best return for the money expended determined. These results should be plotted in the manner shown on Fig. 96.

Fig. 97 shows the value of the saving for various kinds of coating according to the experiments of Rietschel.

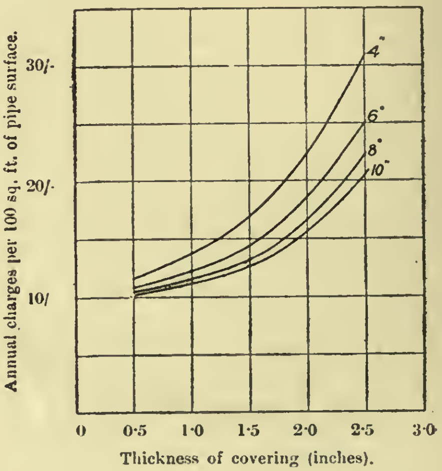


FIG. 96.

The Emission of Heat from Vertically Placed Surfaces.

The only form of vertical surface that we have yet considered is that of a round pipe. By far the greater portion of heating surface used in practice, consists of sectional radiators.

As most radiators consist of extended surfaces placed in close proximity to one another, it will be well to consider generally the effect of this juxtaposition on the total heat emission.

Rietschel describes an experiment on two smooth parallel plates 6′ 6″ high, and 27½″ wide, heated by steam with a difference of 146° between the temperature of the steam and the temperature of the air.

These surfaces were placed opposite one another at different distances apart.

It was found that there was a reduction of the total heat emitted compared with the same surfaces freely exposed to air of the following amounts :—

Where the distance apart was	⅝″	the reduction was		82%
,, ,, ,,	1¾″	,,	,,	58%
,, ,, ,,	4″	,,	,,	51%
,, ,, ,,	8″	,,	,,	43%
,, ,, ,,	20″	,,	,,	34%

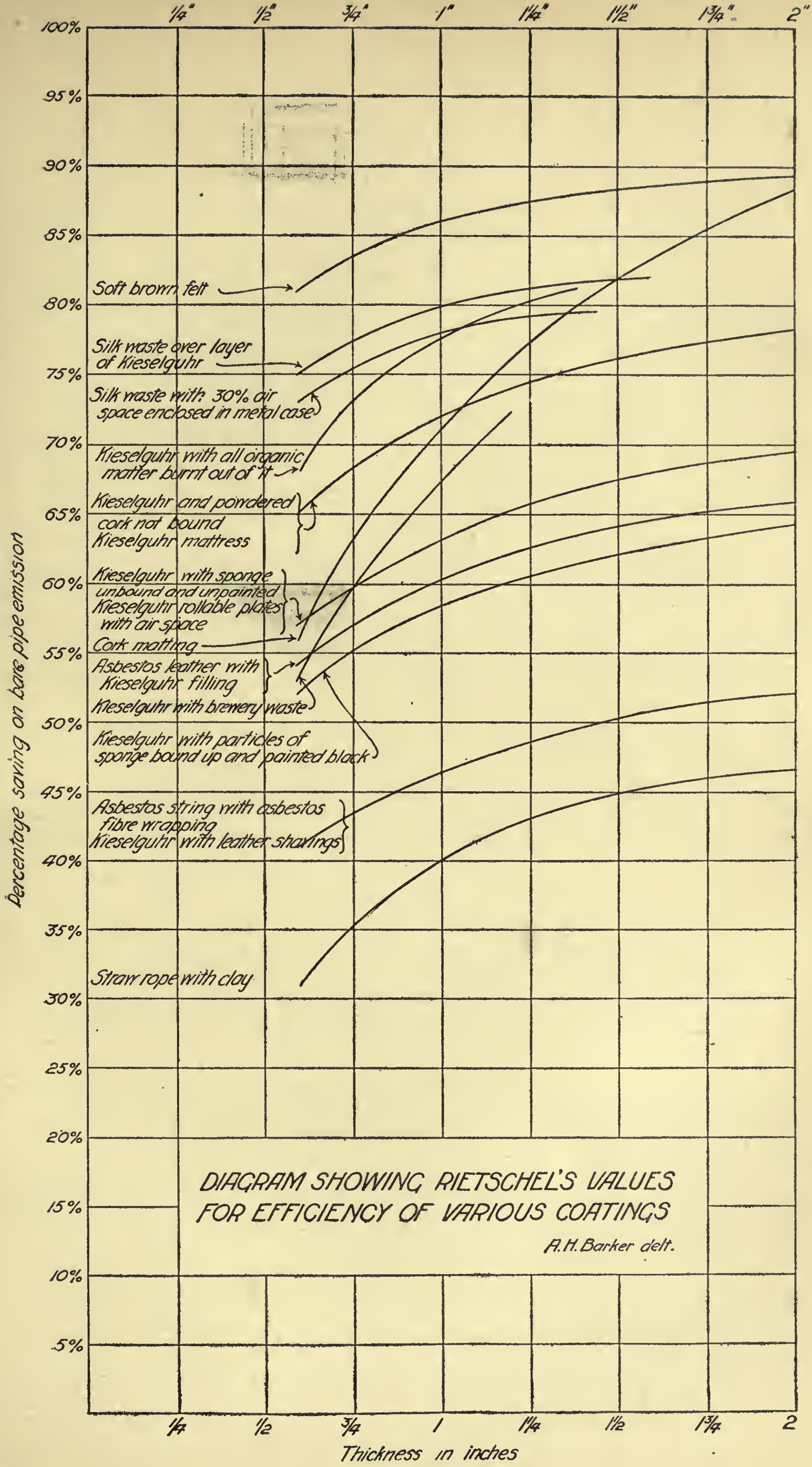

1/4"
1/2"
3/4"
1"
1 1/4"
1 1/2"
1 3/4"
2"
100%
95%
90%
85%
80%
75%
70%
65%
60%
55%
50%
45%
40%
35%
30%
25%
20%
15%
10%
5%
Percentage saving on bare pipe emission
Soft brown felt
Silk waste over layer of Kieselguhr
Silk waste with 30% air space enclosed in metal case
Kieselguhr with all organic matter burnt out of it
Kieselguhr and powdered cork not bound
Kieselguhr mattress
Kieselguhr with sponge unbound and unpainted
Kieselguhr rollable plates with air space
Cork matting
Asbestos leather with Kieselguhr filling
Kieselguhr with brewery waste
Kieselguhr with particles of sponge bound up and painted black
Asbestos string with asbestos fibre wrapping
Kieselguhr with leather shavings
Straw rope with clay
DIAGRAM SHOWING RIETSCHEL'S VALUES FOR EFFICIENCY OF VARIOUS COATINGS
A.H. Barker delt.
1/4
1/2
3/4
1
1 1/4
1 1/2
1 3/4
2
Thickness in inches

FIG. 97.

The effect of Fixing Radiators at Different Distances from Vertical Walls.

The heat emission from a radiator fixed in the middle of a room some considerable distance from the wall is obviously different from its emission when placed near to or under the influence of surrounding objects. The heat actually radiated to those near objects has the effect of raising their temperature, and they themselves thereby increase in radiating power. The rays which they emit impinge upon and are absorbed by the radiator, thereby tending to diminish the net loss of heat from it.

Another effect of placing the radiator in close proximity to a wall or vertical surface is that the layer of air between them is raised in temperature, and therefore the loss of heat from the side of the radiator facing the wall or obstacle is further reduced.

But there is another and precisely opposite effect. The raising of the temperature of the air causes the space between the radiator and the wall to act as a kind of upcast flue, whereby a constant and considerably increased velocity of air is maintained over the surface of the radiator, having the effect not only of tending to keep the temperature of the space down, but also when the flue has the most favourable proportions of even increasing the heat emission over what it would be if fixed in the middle of a room.

A flat radiating plate 3′ 3″ high was placed in different situations with regard to a wall by Rietschel, and results observed when placed at different distances from a wall, the heat given off was *increased* as follows :—

0.8″	by	4%
5′ 5″	,,	15%
10″	,,	10%

over the same surface fixed in the middle of a room.

With a flat radiating surface, 6′ 6″ high, a diminution of heat emission of 18% was observed when the distance from the wall is .8 of an inch.

When the distance is 5.5″ an *increase* of 6% was observed, and
,, ,, 10″ ,, 13% ,,

The difference in fact depends jointly on the increase in the velocity of air and on the increase in the temperature in the space. Where the former has a greater influence than the latter there is an increase in emission, and vice versa.

The heat emission from a radiator is increased if it is placed opposite a window surface. The differences due to this cause are said to amount to as much as 7% of the total in extreme cases.

Casing Radiators.

The concealment of a radiator by perforated casing has a variable effect, according to the character of the facilities provided for the circulation of air over the radiator. If surrounded by such a casing that the space around the radiator forms an efficient air flue, the heat emission is not diminished.

For instance, if a free air space is left in the bottom of the casing and a free open-work grating, or wire mesh, fixed over the top so that the air has a free course over the radiator, the heat emission may be taken as being the same as if no casing existed.

When, however, a solid shelf is fixed over the radiator, so that the condition of free circulation does not obtain, the heat emission may be largely reduced.

The common practice is to reckon for such casing a diminution of heat emission of 25%.

To the same category belongs the unfortunately common and filthy practice of fixing heating pipes in trenches below the floor in churches or other buildings, covering them up with gratings. In this case it is a common practice to reckon the loss of heat emission as 30%, but according to the experience of the Author the figure 50% would be nearer the mark.

Heat Emission from Radiators.

The form of radiator which is used in a very large proportion of modern heating installations consists of a freely exposed vertical sectional two column design, which can be supplied of any desired number of sections.

When radiators were first introduced they assumed a wide variety of different designs, which during the past few years have been gradually eliminated, until those commonly employed are at the present time reduced to about half a dozen different well-defined types. Each of these types has, of course, its own efficiency, and it is impossible to give figures for these other than the most ordinary type.

Influence of Height of Radiator.

According to Rietschel the heat emission per square foot of radiating surface is not greatly altered by the increase in height, so long as that height does not exceed about 3′ 6″. Above this, however, the rate of emission decreases, and it is customary to decrease the allowance for radiators above 3 feet high by 10%. This is a more or less arbitrary figure, which roughly approximates to accuracy for the particular case where the height is 45″.

The reason for this is, of course, that the increase of velocity of the air passing over the upper part of the radiator compensates for the loss due to increased temperatures of the air in contact with the surface, above 3 feet high the compensation is not complete.

In cases, however, where the form of the radiating surface is such that the increase in the velocity of the air is prevented by irregularities of the surface, such, for instance, as in the case of ribbed radiators, the addition to the height is attended by a diminution in heat emission per square foot.

These remarks, of course, do not apply to the case where the velocity of the air is produced by mechanical means.

A greater increase of heat transmission is produced by forcing the velocity of the air, as will be seen in the section devoted to that branch.

The experiments of the National Radiator Co., however, give the following proportion between the value of *K* for radiators of different heights, the design being otherwise the same (single column):—

Height of radiator	23″	26″	32″	38″
Relative value of *K*, 32″ = unity ...	1.14	1.06	1.00	1.02

Artificial Increase of Surface by Cast Projections.

In regard to the extent to which the emission from a radiator may be increased by artificial additions to the surface by means of projections, ribs, etc., cast on, it is found that while an increase of the total emission of heat is produced by the addition of ribs to a certain extent, yet where the ribs are greater than about 1½″ they produce an actual decrease of the transmission of heat *per pound of metal,* and therefore ribs above this height are uneconomical from every point of view, especially as the addition of ribs increases the difficulty and danger of casting. The addition of projecting conical pins does in fact largely increase the emission.

Attempts have been made to increase the efficiency of a radiator by fixing vertical air tubes through the exterior tube, but these have not met with considerable success, because the greater increase in cost of this construction is not compensated for by the increased heat transmission, especially as the form of interior cylinder is not favourable to free heat emission. Any heat *radiated* from the interior surface is, of course, absorbed again on the opposite side of the tube, and the high temperature of the air in the tube is such that the emission of heat per square foot by conduction and convection is very small.

Effect of Number of Sections.

The actual conditions of temperature of the water or steam within, and the air without the heating surface, have already been considered in Chapter XII.

Consider a radiator consisting of one single section fixed in the middle of a room. The temperature of the air *surrounding* the radiator on all sides (but not that of the layer in actual contact with it) is practically, in this case, the temperature of the room air, and such a section is under conditions not differing greatly from the case of a vertical pipe of similar height.

It is not possible to specify exactly what is the temperature of the air *in actual contact* with the surface. It is higher above and lower below. The value of K for this case is therefore approximately the same as for a single vertical pipe.

The addition of one section to this single section is clearly equivalent to the addition of inner surfaces only. The two ends of this small radiator may be considered as jointly forming the one single section, for which the value of K has already been considered.

Further, the value of K for a two-section radiator is less than for one-section radiator, because the two inner surfaces of the two sections are not in so favourable a condition for parting with their heat as in the case of the single section.

The value of K for this case is therefore lower than for one section.

The addition of yet another section is, in fact, the addition of two more interior surfaces. So that the longer the radiator becomes, the more nearly does the value of K approach the value for the interior sections only.

The formula, therefore, for the heat emission from a radiator of n sections, of which the temperature is given, is therefore the following form :—

$$a + b\,(n - 1)$$

in which a is the total heat emission in B.T.U. for the two outside surfaces, and b that for a pair of inner surfaces.

The further addition of each section, since it merely adds in effect two inner surfaces to the radiator, corresponds to an addition of b B.T.U.

If each section contains c square feet, then

$\frac{c}{a}$ is the number of B.T.U. given off from each square foot of outer surface.

$\frac{b}{c}$ the corresponding figure for inner surfaces.

For ordinary type of sectional radiator Rietschel gives the value of $\frac{a}{c}$ as follows, varying with the temperature difference, where the difference of temperature between room air and water is

	Actual temp. of the radiator when the room is 60°.	Value of $\frac{a}{c}$	and $\frac{b}{c}$
Under 72°	132	1.68	1.21
72— 90°	132—150	1.78	1.31
90—108°	150—168	1.80	1.38
108—126°	168—186	1.82	1.43
126—144°	186—204	1.84	1.44
Over 144°	over 204	1.86	1.46

For low pressure steam radiator

$$\frac{a}{c} \text{ is } 2.50 \text{ and } \frac{b}{c} \text{ is } 1.67$$

For high pressure steam (over 10 lbs.)

$$\frac{a}{c} \text{ is } 2.57 \text{ and } \frac{b}{c} \text{ is } 1.73$$

For very long radiators the temperature of the air surrounding the middle part is higher than with a shorter radiator, so that the above formula is not quite accurate. The emission of a radiator 20 sections long is about 5% to 8% less per square foot than of a 10 section at the same temperature.

Distance Apart of Sections.

Some years ago it was the custom to make the radiators so close together that the space between the sections could not be reached otherwise than by a brush or other implement. The results of further accurate experiments in heat emission, combined with modern notions of hygiene, however, dictate that the space should be greater.

For reasons of hygiene it should be possible to reach all parts of a radiator by hand for dusting purposes, and this alone dictates the condition that the minimum distance between two opposite surfaces of a radiator should be 1″, and almost all radiators of plain sectional type are now made with this space between. For hospital purposes the space is greater.

The efficiency of a radiator with greater spaces is greater than with less spaces. The effect on the value of K of increasing the amount of air space is shown in the table below, standard radiation having sections fixed 3″ apart being taken as unity.

Distance cent. to cent.	2½″	3″	3.75″	4″
Relative value of K	0.945	1.00	1.10	1.12

Effect of Temperature of Media.

According to the theory of the subject the heat emission is proportional to the difference of temperature between the mean temperature of the medium in the radiator and the mean temperature of the surrounding air.

It has already been pointed out that this theory is not accurate, it is not even approximately accurate, and its inaccuracy is compensated for in practice by giving different values of the co-efficient *K*, corresponding to different temperatures.

The result of this variation in the values of *K* is that the heat emission corresponding to different temperatures is not straight, but slightly curved, according in the manner shown on the diagram, Fig. 98.

This curvature is not sufficiently great to be of any considerable moment.

Actual Heat Emission from Radiators.

The heat emitted from a radiator can only be determined by prolonged and careful tests. It is very difficult to give results which will be applicable to all kinds and sizes of radiators. As previously explained, every design has its own particular efficiency, not only so, the heat emitted per square foot, from radiators of the same design, but of different size varies considerably. The total emission at a constant temperature is not in proportion to the surface of radiator. General figures covering all radiators cannot therefore be obtained.

A classical series of tests on radiator emissions were made by Rietschel some years ago, and the results are shown on Fig. 98. These apply accurately only to the form of radiator then in use. Since that date considerable modifications and improvements have been made in radiators, and it is probable that these figures are about 15% to 20% too low for the modern type of radiator. Correct results can only be obtained for any particular type by a special test, and it has already been pointed out that the results secured for one particular radiator vary considerably, according to the conditions in which it is placed. The subject is therefore one of exceptional difficulty.

The results of Rietschel's tests are embodied in his co-efficients, a table of values of which is given below. These values are shown plotted on the diagram, Fig. 98. It will be seen that owing to the arbitrary limits adopted in the table, the points do not lie on any regular curve, and that a sudden jump in the value of the co-efficient is given at particular temperatures.

It appears obvious that an arbitrary jump of this kind cannot represent accurately the results of tests. These values, therefore, have been interpolated by the Author, and a series of mean curves drawn between the points representing Rietschel's values, and all calculations are made from the co-efficients obtained from the adopted curves.

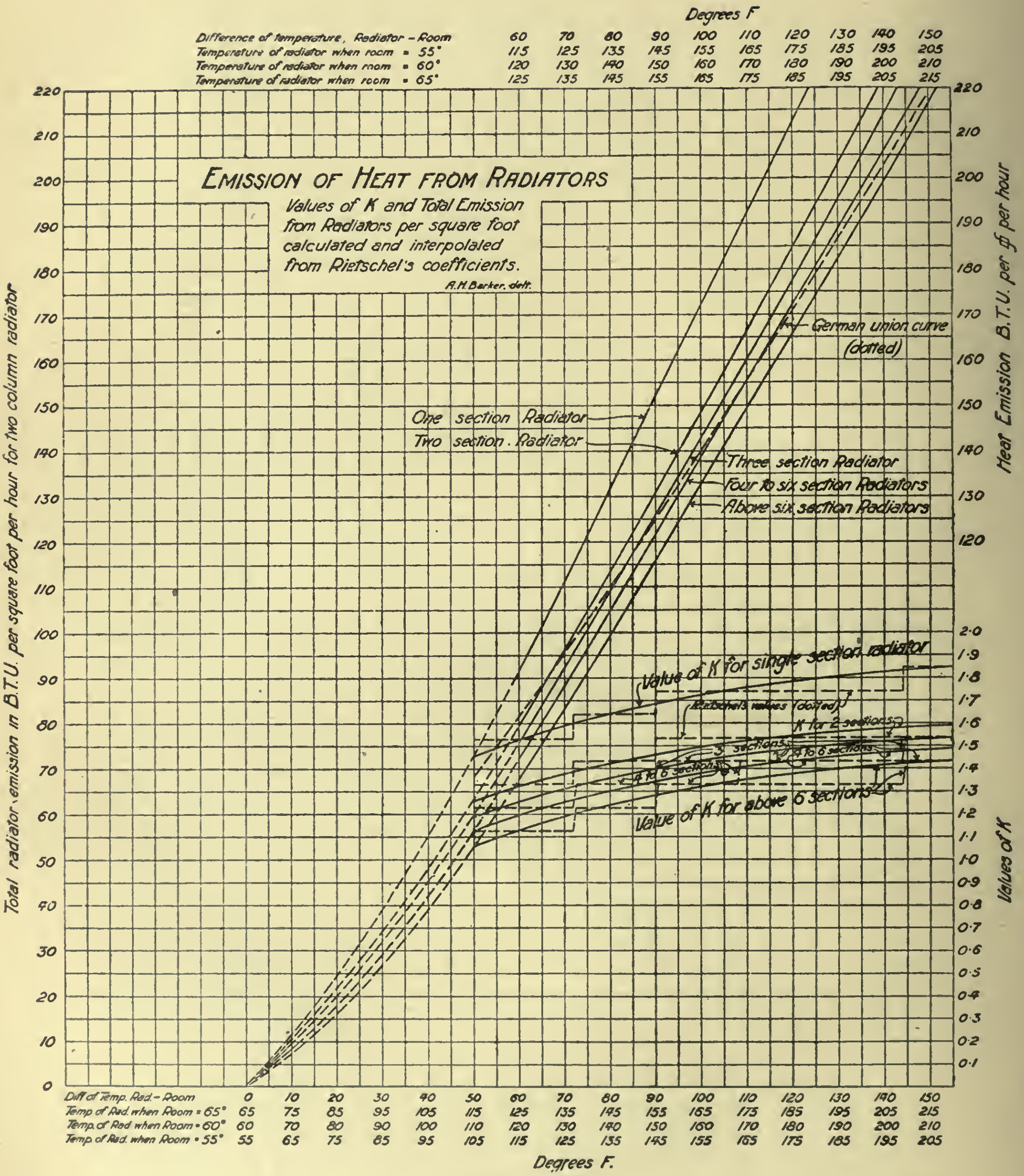

Degrees F
Difference of temperature, Radiator – Room 60 70 80 90 100 110 120 130 140 150
Temperature of radiator when room = 55° 115 125 135 145 155 165 175 185 195 205
Temperature of radiator when room = 60° 120 130 140 150 160 170 180 190 200 210
Temperature of radiator when room = 65° 125 135 145 155 165 175 185 195 205 215
Emission of Heat from Radiators
Values of K and Total Emission from Radiators per square foot calculated and interpolated from Rietschel's coefficients.
A. H. Barker. delt.
German union curve (dotted)
One section Radiator
Two section Radiator
Three section Radiator
Four to six section Radiators
Above six section Radiators
Value of K for single section radiator
Rietschel's values (dotted)
K for 2 sections
3 sections
4 to 6 sections
4 to 6 sections
Value of K for above 6 sections
Total radiator emission in B.T.U. per square foot per hour for two column radiator
0 10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200 210 220
Heat Emission B.T.U. per ☐' per hour
120 130 140 150 160 170 180 190 200 210 220
Values of K
0·1 0·2 0·3 0·4 0·5 0·6 0·7 0·8 0·9 1·0 1·1 1·2 1·3 1·4 1·5 1·6 1·7 1·8 1·9 2·0
Diff of Temp. Rad. – Room 0 10 20 30 40 50 60 70 80 90 100 110 120 130 140 150
Temp of Rad. when Room = 65° 65 75 85 95 105 115 125 135 145 155 165 175 185 195 205 215
Temp of Rad. when Room = 60° 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200 210
Temp of Rad. when Room = 55° 55 65 75 85 95 105 115 125 135 145 155 165 175 185 195 205
Degrees F.

FIG. 98.

The experiments were made on ordinary two column radiation with a 1″ air space between the sections, corresponding to 3″ centres between the sections.

Other tests made since that time appear to indicate that the values adopted by Rietschel are too low, this result being probably due to improvement in the design of radiators.

The curves of total emission, given on Fig. 98, are calculated from the values of *K* so determined, and Table XVII. has been drawn up showing the size of radiator required at different mean temperatures, in order to emit varying quantities of heat. The amount of heat actually emitted by a radiator varies considerably, according to the method of connection adopted. It will be clear, therefore, that there is a considerable element of uncertainty about the correct figure to be taken for the total emission of a radiator, much more so than in the case of a heating pipe.

The German Union of Heating Engineers have adopted a table which approximates closely to the values as determined by Rietschel. This table may be taken to represent the average allowance in practice, and is almost universally adopted as a standard on the Continent. The table is given in a somewhat different form from the tables of Rietschel.

In order that the two standards may be compared, the curve corresponding to the German Union figures is also plotted on the same diagram. The form in which the results are given by the German Union is slightly different from that given in the figure. Here there are four different standard room temperatures assumed, corresponding to 50°, 59°, 68°, 77°, and the mean of the radiator is given for each of these temperatures, with given graduated values of the flow and return. These results may be understood from an investigation of the diagram alluded to.

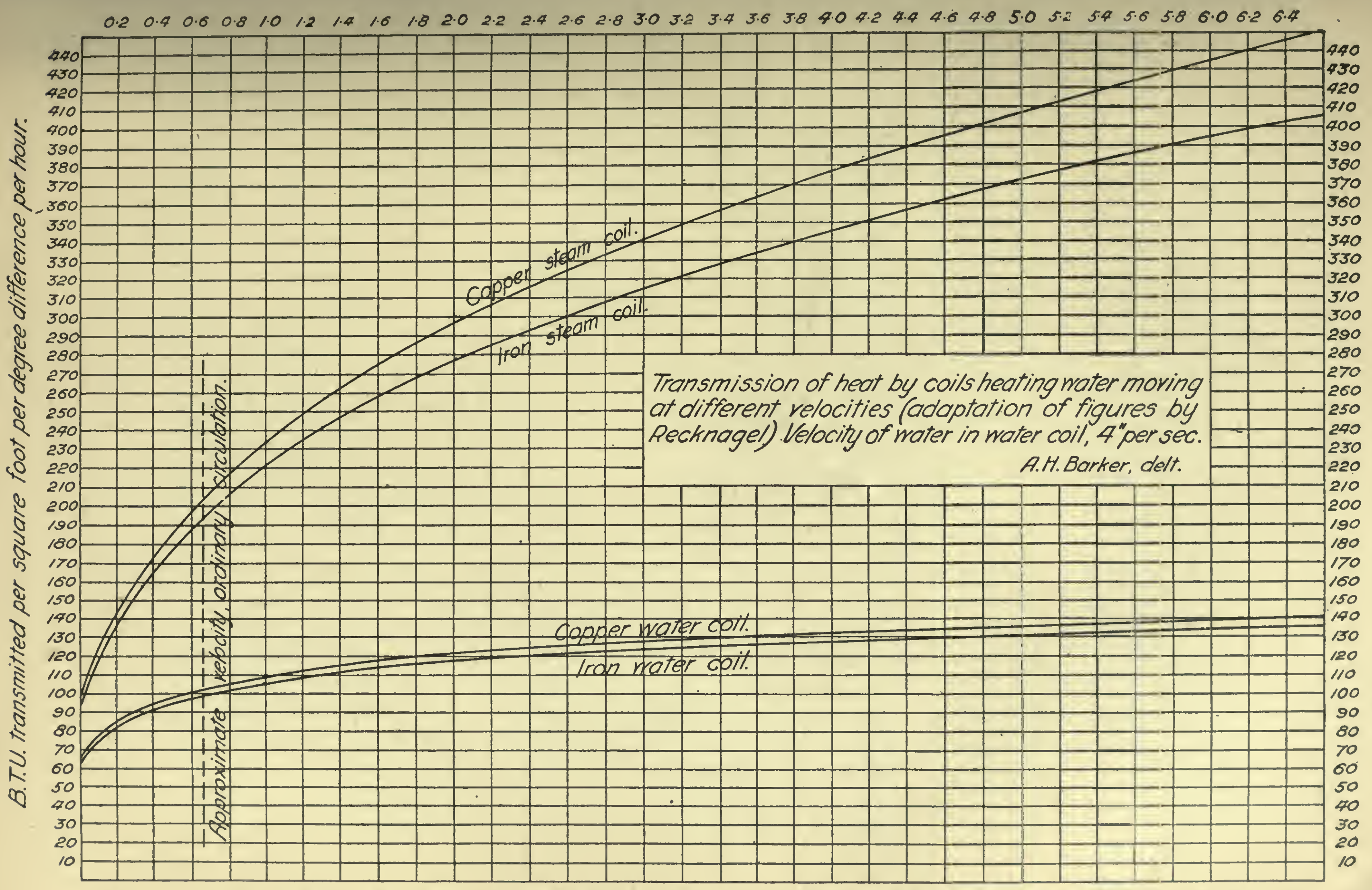
It will be seen from the table given by the German Union that a flow temperature in the radiator less than about 175° F. is not contemplated, and that the heat emission is taken to follow a straight line—a law which is approximately, but not accurately correct.

The table, No. XVII., has been calculated in what the Author believes to be the most convenient form for practical use. In this table the heat emission for radiators of various sizes is given for all temperatures of flow and return, with intervals of 5°. The minimum temperature difference contemplated between the flow and return is 20° for the upper ranges of temperature and 15° for the lower ranges.

Three standard room temperatures have been assumed, namely, 55°, 60°, and 65°, which will cover the great majority of cases in this country.

The corresponding curves of the heat emission from pipes of various sizes has been calculated per foot run of pipe, also from the experiments of Rietschel. The temperature basis here is also the mid temperature.

Comparison is also given with the tables derived from the experiments



Transmission of heat by coils heating water moving at different velocities (adaptation of figures by Recknagel) Velocity of water in water coil, 4" per sec.

A.H. Barker, delt.

FIG. 99.

of the National Radiator Company. All the figures given in the tables are for standard conditions only, namely, still air and normally warm surroundings. Any alterations in these figures can be calculated in the form of a percentage from the particulars given in the previous parts of the present chapter.

A table is also given of the values of K for irregular and unusual kinds of heating surface which are less commonly employed than plain pipes and radiators.

Emission of Heat from Steam Heated Radiators.

The figures are given for these in the form of tables.

For steam heated surface it is not necessary to give a number of different values of K, since the value actually adopted is constant in practice, although, as has been explained, the actual heat emission varies to some considerable extent with the amount of air in the radiators. This is not a factor of which account can be taken. It is therefore disregarded in calculations, and a somewhat low value of K adopted to suit average conditions.

All steam heated surface, therefore, is assumed at one given temperature of the room. A slightly enhanced value, however, is given for high pressure steam (Table XIX.).

High Pressure Hot Water.

A table of co-efficients is also given for this, for mid temperatures up to 270° F. (Table XX.).

Effects of the Increase of Velocity of the Fluids.

Figures showing the effect on the total heat transmission of various velocities of air passing over the surface are given in Chapters XXXI. and XXXII. The standard form of heating surface for this class of work has hitherto been a number of parallel rows of wrought iron pipe about 1.3″ in outside diameter, and about 1½″ centres, leaving slightly under a ¼″ space between the pipes, the part of the air being perpendicular to the axis of the pipes.

Radiator tubes and special forms of heating batteries are now being increasingly employed for the same purpose.

Fig. 99 shows the variations due to the velocity of the water in the rate of transmission from steam to water and from hot water to cooler water through tubes of iron and copper.

CHAPTER XVII.

ON THE TEMPERATURE MAINTAINED IN A BUILDING.

Meaning of the Thermometer Reading.

IT is desirable to acquire a perfectly clear conception of the meaning of the expression "temperature of a room." Strictly speaking, any accurate thermometer registers *its own temperature only*. This is only the same as that of objects in its neighbourhood, in so far as the thermometer has assumed the temperature of its surroundings. Thus, *the temperature registered by a thermometer is not necessarily that of the air surrounding it.*

Since all the objects in a room are under the same influences there is no reason why any thermometer should assume a temperature different from that which the furniture or other objects in the room will ultimately attain. The thermometer, being a small object, takes up the temperature more rapidly than do larger objects. In fact the reading of a thermometer may be taken to be a sample of the temperature which the furniture in the room has or will ultimately attain.

If a thermometer is hung in any chamber exhausted of air having walls at a particular temperature, it must ultimately attain the same temperature as the walls, on account of the mutual exchange of radiation between the thermometer bulb and the walls, as explained in Chapter XII.

If into this imaginary vacuous space a certain quantity of air is introduced at a higher temperature than the walls, the effect on the thermometer will be the resultant effect of the interchange of radiation between the bulb and the cold walls on the one hand, and of the interchange of heat by convection currents between the warmer air and the bulb on the other. The reading will reach a point intermediate between the temperature of the surface of the walls and that of the air, such that the amount of heat communicated *to* the bulb by the warm air is exactly equal to the loss of heat *from* it in consequence of the interchange of radiation with the cold walls. A thermometer in such circumstances neither registers the temperature of the air nor that of the walls, but assumes some reading intermediate between the two.

Thus, in a cold building, the air of which has been rapidly heated above the wall temperature by forcing the heating apparatus, the temperature of a thermometer suspended in the middle of the building normally reads some degrees lower than the actual temperature of the surrounding air.

Next consider the case where the walls of the chamber are at a higher temperature than the introduced air.

In these circumstances the thermometer will read a temperature lower than that of the walls on account of the influence of the cooler air, but its reading will be higher than that of the air, to such an extent that the amount of heat received by the thermometer as the result of the mutual interchange of radiation with the walls will be just sufficient to provide for the loss of heat by the thermometer bulb, due to its being surrounded by air at a lower temperature than itself.

Consider the effect of blackening the bulb of the thermometer. The effect of this is to cause the bulb to absorb the whole of the radiation which falls upon it. It will reflect none. It will also increase the rate of emission of heat in the form of radiation from the thermometer bulb.

On the other hand, the blackening of the bulb will not appreciably alter the rate of loss of heat by convection from it to the *air* surrounding it unless the entire thermometer is enclosed in an outer vacuous bulb, as are all radiant heat thermometers.

There will be no difference between the readings of a black and a bright thermometer when the whole space is vacuous, for in this case, as has been previously explained, the temperatures will assume the same value on account of the fact that the radiating and reflecting power of any surface vary inversely.

The result is that the blackening of the bulb will only cause a rise in the temperature of the thermometer when the air is at a lower temperature than hot bodies radiating heat in the same room. The temperature assumed by any body in such a room depends on its position with regard to the radiating surface. If it is so situated as to receive and absorb the radiation it rises to a higher temperature than if not so situated.

Means for Determining the Intensity of the Radiant Heat.

If the blackened thermometer is surrounded by a vacuous bulb it is shielded from the effect of direct contact of air and will show a temperature influenced by radiant heat only.

It will thus be understood that the reading of an ordinary thermometer hung in the room does not of necessity indicate anything very definite except its own temperature, and probably also the general temperature of the solid contents of the room, depending as it does, not only on the temperature of the air, but also on the gross amount of radiant heat absorbed by the bulb.

To surround the bulb with a plain hollow cylinder of metal does not produce any considerable alteration of the reading, for this cylinder soon attains the same temperature as the bulb itself would have attained if the cylinder had not been there.

Means for Finding Temperature of the Air.

In order to indicate accurately the temperature of the air, a thermometer must be entirely shielded from *all* radiation except that proceeding from surfaces at the same temperature as the air. To effect this, special appliances must be provided for ensuring that the temperature of all surfaces surrounding the thermometer is identical with the air temperature. This effect can only be secured by some such appliance as the following :—

Let the thermometer bulb be surrounded by a small hollow double cylinder containing water in the annular space between the two cylinders. The temperature of the water can be regulated by the admission to the annular space of warm or cold water as may be necessary, a stirring rod of appropriate shape being provided to ensure the uniformity of temperature in the water. The thermometer is hung in the space inside the inner cylinder. A delicate apparatus is provided for the indication of movements of air up or down in this cylinder. Such an appliance may be a very light anemometer vane suspended by a silk thread, which will rotate in one direction or the other, according as the air passes up or down.

When, therefore, the temperature of the water in the hollow space is above the temperature of the surrounding air, this will cause an up current through the hollow space, which will be indicated by the rotation in one direction of the indicator.

On the other hand, if the water in the hollow space is at a lower temperature than the air, a downward current will be caused, which will cause the rotation of the indicator in the opposite direction. If then the temperature of the water in the cylinder be exactly adjusted until there is no movement of the air up or down within the central space, it is certain that a thermometer is surrounded not only with air at the same temperature as the body of the air in the room, but also that all the radiation which reaches the thermometer bulb proceeds from a surface at the same temperature. Only in these conditions will a thermometer read exactly the temperature of the air surrounding it.

Effect of Furniture in a Room.

Compare the readings of a thermometer bulb in a room warmed by radiant heat only, of which the walls are cold, in two cases. In the first the room is supposed to be empty, and in the second to be filled with furniture or other light articles, which are readily affected in temperature by the radiation passing through the room.

In the first case the temperature of the thermometer will vary from that of the air only, on account of the presence of the cold walls. This effect will be much less in the second case, because the furniture and other light articles within the room soon absorb the radiant heat and will themselves, by the absorption of heat rays, rapidly assume a higher temperature than the walls. The surfaces of the latter are kept cool by conduction to the outer air, those of the former are not. Those pieces of furniture which are nearest to the radiating body will acquire a higher temperature than those further away. They will therefore speedily become objects radiating heat to the thermometer. The presence of their relatively warm surfaces will produce a notable effect on the temperature of the thermometer. They will in fact trap and retain as heat within the room the radiant energy which would otherwise reach the walls or the window and so be lost.

By their presence the amount of radiant heat affecting the thermometer will be increased. This appears to be an explanation of the fact frequently noticed in practical life, that a furnished room is always warmer than an empty room when the same amount of heat is being supplied to it.

This effect is not necessarily due to the fact that *the air* is at a higher temperature, though this may also be true, but essentially to the fact that the effect on the thermometer is different.

These facts should be thoroughly understood by the heating engineer.

Review of Validity of Calculations.

It is desirable to review as a whole the calculations on which the engineer is commonly accustomed to base his work in order to acquire a clear conception of what they profess to be, what they really are, and the extent to which they are available for enabling the heating effect of an apparatus to be foretold.

In the first place it must always be remembered that the estimations of heat loss (which form the greater part of the calculations relating to a heating apparatus) profess to give the transmission of heat *in certain assumed conditions only*. If these conditions are not fulfilled, the result of the calculation does not even profess to represent the transmission of heat. In this country these assumed conditions are—

(1) That the outside temperature is constant, generally 32°.
(2) That the air is still and dry.
(3) That the supply of heat to the interior of the building has been maintained for so long a time that a steady temperature has

been attained in all parts of the room, and a constant flow of heat established.

(4) That the rate of interchange of air has the given value.

It has been shown that these conditions are practically never maintained in practice, so that the calculated heat loss is an ideal quantity only. It remains for us to inquire what is the relation of this ideal quantity to a quantity actually lost, in other words, to what extent these calculations are available for enabling the effect of the heating to be foretold in a practical case in any given conditions.

It is an elementary principle of technical or any other science that if it is to be possible in any circumstances to foretell the result of any physical experiment of any kind, the value and effect of all the factors which have an important influence on the result must be known and allowed for with just so much exactitude that the deviation from actual accuracy in the value assigned to any factor does not produce any important vitiation of the accuracy of the result.

If all the factors are not so known or not properly allowed for the result of the experiment cannot be determined.

Where the result aimed at is the temperature of a thermometer in a building, we have, therefore, to review the calculations as a whole to see whether they fulfil this elementary requirement.

The ultimate criterion of practical utility in these calculations, however, is not whether the co-efficients are correct in the absolute or scientific sense, but primarily whether the method and principles on which they are founded is valid and correct, and secondly, whether the values adopted for the various co-efficients bear a correct proportion to one another. The latter is certainly a more important question than the absolute accuracy of any one or more of the factors. The first of these questions resolves itself into this :—

Is the heat required for any given duty really proportional to the sum of certain respective fractions of the area of the wall, roof, floor, window, etc.?

The second question resolves itself into this :—

If a large number of rooms of different shapes, situations, sizes, are carefully measured, and if the ordinary co-efficients are applied to each so as to determine the number of B.T.U. required to maintain each at the standard temperature, then if the respective calculated numbers of B.T.U. are actually supplied to each of these rooms, will the temperature maintained be the same in all cases?

If there is variation in the result, it is a positive proof that the proportions between the co-efficients is incorrect. If the temperatures are in all cases the same, but different from the calculated standard temperature, it shows that the absolute values of the co-efficients are wrong, although the proportions are correct.

There is a good deal of practical evidence in existence to show that the value of these co-efficients needs to be revised to suit the materials and conditions in this country. The co-efficients now in practical use are based on experiments made many years ago by Peclet, in France, and have never been thoroughly checked or examined, at any rate, for many years.

Instances are known to every careful observer where very careful calculations have failed to secure that uniformity of result which is to be desired. Although this bare fact may be apparent it is always a difficulty that it is rarely possible for the practitioner, even if he has the analytical ability required, to devote the time to the carrying out of the very elaborate experiments which would be necessary to investigate closely the cause of the discrepancies between calculation and result.

The absolute accuracy of the co-efficients is of less importance than their relative accuracy, because any variation from absolute accuracy may be compensated for in practice by increasing the temperature of the installation throughout. Errors in proportional accuracy are apt to cause very serious trouble.

In considering the extent to which the calculations may be relied on to enable a temperature result to be predicted, it has been explained in the previous chapter that the temperature maintained in a room by a given supply of heat depends in a very important degree on the volume of cold air which passes through the room during every hour, and it must again be distinctly pointed out that no means whatever exist for making any reliable calculations to determine how much air will pass through any room in an hour, except in cases where the room is provided with flues of definite height or other well defined means for ventilating, and even in these cases the result of experimental observations is not infrequently widely different from that of calculation for reasons which have already been thoroughly discussed.

However accurate, therefore, the co-efficients of transmission may be in the assumed conditions, it is clearly impossible to make the total calculation accurate. It is well known, too, that the heat lost from a building depends very largely on the external velocity of the wind. With a given outside temperature, the temperature maintained in a room may vary 5° or 8°, according to the strength and direction of the prevailing wind, partly on account of the additional air interchange produced, and partly owing to its effect on the temperature gradient in the walls.

It is thus apparent that natural causes alone will prevent the temperature maintained at any one time or on any one day in any given building from being accurately foretold, and that such a temperature cannot furnish a reliable test of the power of the heating apparatus.

Interruptions of Heating.

Apart, however, from these variable natural influences, there are features in the practical working of a heating apparatus for which orthodox practice makes no regular allowance, and which certainly have a large influence on the result secured.

It has been shown and can easily be proved by observations on any existing heating apparatus, that the temperature actually maintained in any building with any given and absolutely constant outside conditions, depends on the length of time the heating apparatus has been continually at work, and on the question whether or not the heating has been interrupted during the night and for how long.

When an apparatus is kept uniformly and continually at work, the temperature attained will continue to rise for many days or even weeks. If, therefore, no proper allowance is made for the interruption of heating in the calculations the "internal temperature" which forms the objective of the whole calculation will never be ultimately attained. Any temperature other than that ultimately attained by prolonged and continuous heating with a constant outside temperature is a variable value which can be raised or lowered to any degree within certain limits, according to the length of the period of interruption and according to the variations of the outside temperature.

If the calculated result is correct for continuous heating, it cannot conceivably be correct for practical work in which the heat is always discontinuous. If then the calculated temperature has to be attained or maintained with discontinuous heating, either the discontinuity must be allowed for by working the apparatus at higher power than has been calculated for, or the power of the apparatus must be increased.

It must always be remembered that if the walls are heated up by the apparatus to a certain point during the day, they will continue to lose heat to the outer air at almost exactly the same rate throughout the night The rate of loss only diminishes slightly during the period of interruption, owing to the large mass of heat retained in the walls.

If, therefore, heat is only supplied to the room during the day, the amount supplied during the day must be sufficient to provide for the loss that takes place during the night as well as during the day. The orthodox method of calculating for heat losses either leaves this essential and important fact out of sight or attempts in a somewhat feeble manner to provide for it by the addition of an arbitrary percentage. This addition is a proof that so far as the co-efficients employed are found to be satisfactory in practice they are essentially based on the results of practical experience, and not on their correctness from the absolute or scientific standpoint.

Indeed, the fundamental assumption underlying all these calculations is not that the co-efficients themselves are absolutely correct, but that the basic calculation furnishes a reliable jumping off place or starting point for an estimate of, or as it really is, a guess at the amount of heat which will in practice be found satisfactory. Fundamentally, it is not a question of scientific accuracy, but of practical adequacy.

The fundamental assumption is that if the apparatus is of such a power as would in the assumed conditions, and with the assumed co-efficients, maintain a certain calculated temperature, then it will be of sufficient power for practical use, so as not to cause its owner to complain.

It will be seen, too, that as the criterion of satisfactory performance of a contract is one of temperature, which temperature in practice is a variable factor, the personal or subjective element must weigh largely in determining whether or not the apparatus is satisfactory. This aspect of the matter will be treated more in detail later in the present chapter.

Relation Between the Internal and External Temperature.

The temperature maintained in a room at any given time has only a very small and indirect relation with the external temperature existing at the same time, especially when the amount of interchange of air is small. It has been shown on page 304 that the factor which directly determines the mean interior *air* temperature of a room at any moment corresponding to a given mean stationary radiator temperature is the actual temperature of the inside surface of the wall at the time. The latter depends chiefly on the heat history of the walls during the previous fortnight or so.

This fact, which has hitherto failed to find recognition adequate to its importance, has a most important bearing on the problem of testing a heating installation.

It has been pointed out that the internal temperature does not follow the external temperature at all closely, except in cases where a comparatively large interchange of air is proceeding between the outside and inside of a room. If there is very little ventilation, variations in external temperature have so small an influence on the internal temperature as to be almost unnoticeable at the time, unless the effect is continued for so long a time that a new rate of heat interchange has had time to become established.

In order to understand this very important fact, assume that a constant external temperature of, say, 30° is maintained round a certain building, in which the ventilation is very small, and that a constant amount of heat is supplied to the interior, as described on page 306, exactly sufficient to *maintain* in these conditions the desired temperature. It has been explained it would normally take between a week and a fortnight for the temperature to attain its ultimate value. The latter is the only value of the temperature to which the calculations will apply if the co-efficients used are scientifically correct.

Let it be assumed that at the end of this fortnight the external temperature suddenly drops 10°, the constant supply of heat being continued as before. These changes are shown graphically on Fig. 100.

The first effect of the change would be that an increased rate of heat loss from the outside of the building would be set up. The effect of this increased loss from the outside would only be gradually felt on the inside of the building. It would probably be several days before the full effect was felt inside. The ultimate temperature would then stand about as much above the outside temperature as before, that is about 30°.

If, at the end of this period, there was a sudden change of external temperature in the direction of the weather becoming warmer, say a sudden rise in temperature of 20°, the first effect would be that for some time after the change in the weather the outside surface of the wall would commence to *absorb* heat from the outer air. In consequence of the joint effect of the prolonged period of cooling with an outside temperature 20° and of heating in the interior of the building, the outside surface of the wall would have assumed a temperature of perhaps 25°. When, therefore, the outside temperature is suddenly raised to 40° the effect would be that the direction of flow of heat at the outside of the wall would be reversed. Heat would at once commence to flow *inwards* for a time. The rise of temperature in the wall would gradually pass through the wall in an inward direction.

Within the thickness of the wall for some time after the change there would be a section across which no heat was flowing, in either direction—a sort of neutral section, at which there would be no temperature gradient. The accumulation of heat in the wall in the inside of this section would then gradually raise its temperature up to such a point that the outward flow of heat would be re-established. Ultimately the temperature of the wall throughout would adjust itself with a constant temperature gradient adapted to the new conditions. The ultimate effect, if the heating were continued *at the same power* as before, would be that the temperature difference between inside and outside would be almost the same as with a lower outside temperature.

But the *new conditions could only be established after a further prolonged period of heating.* The whole of the substance of the wall would have to be heated to correspond to the new conditions, and *only after this had been done would the new value of the internal temperature prevail.*

Of course, such sudden and precisely defined weather conditions never are obtained in practice. The external temperature changes *gradually*

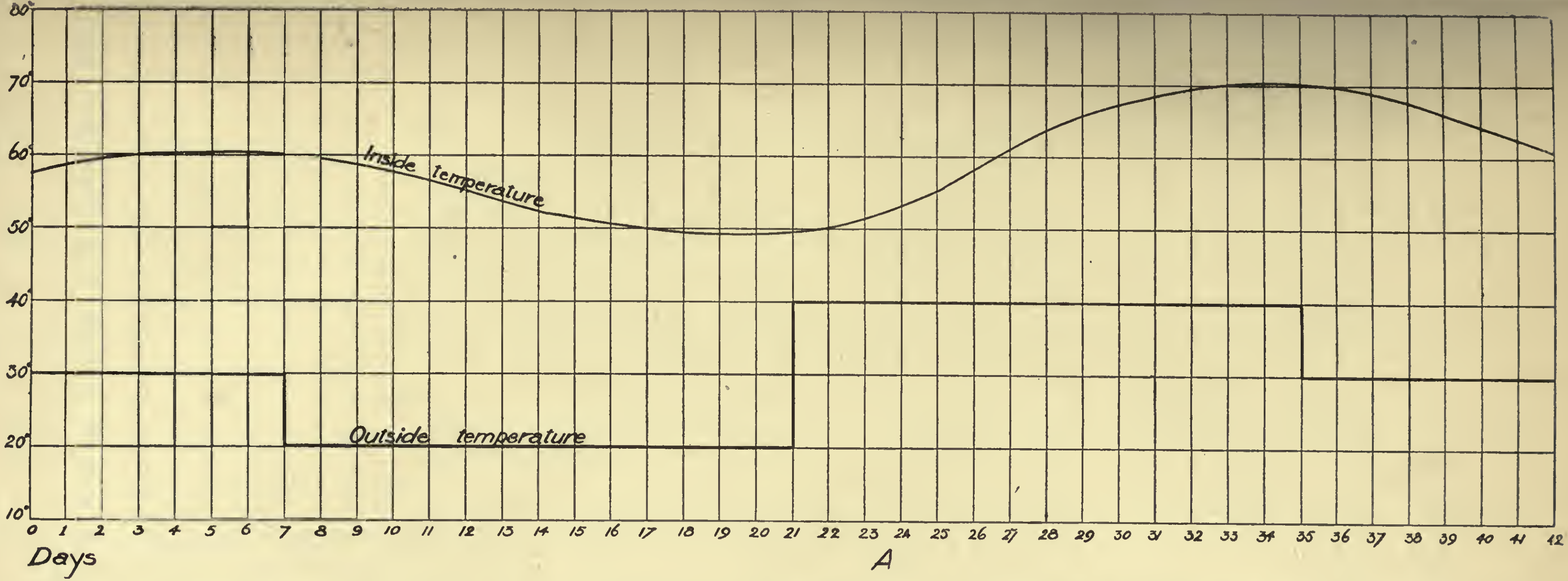


FIG. 100.—Variations in inside temperature in a building with thick walls with constant heat supply and suddenly varying outside temperature.

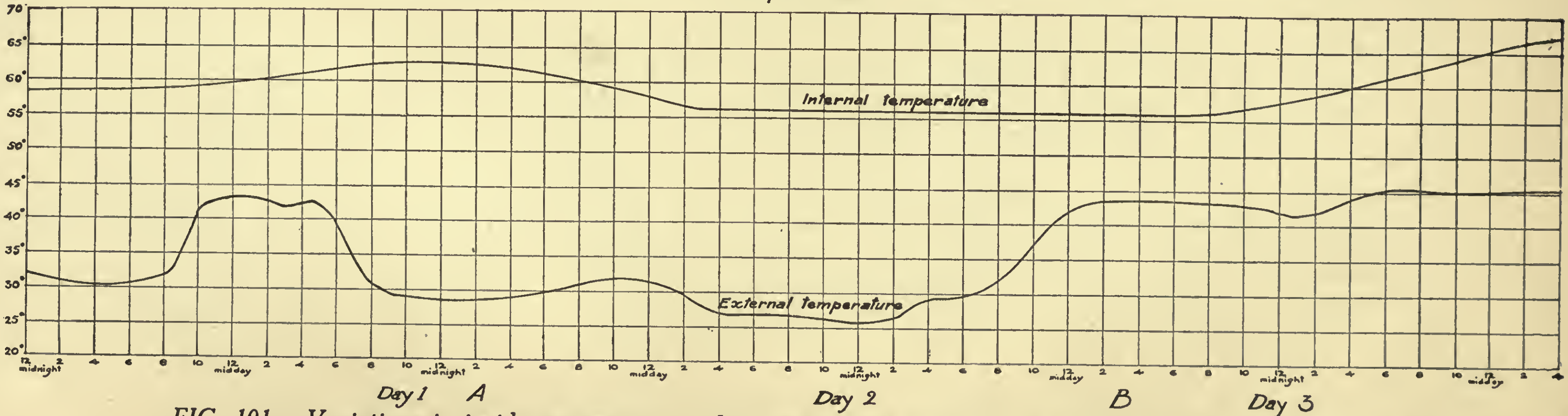


FIG. 101.—Variations in inside temperature with constant heat supply and gradually varying outside temperature.

from one value to another, varying all the time from a high to a low level with the ordinary diurnal variations, and in such conditions it is not possible that any precise law can be laid down showing the relation between the internal temperature at any moment and the external temperature.

The outside temperature will in practice vary in some such manner as shown in Fig. 101, which comprises a period of nearly four whole days. If in these circumstances a constant supply of heat is maintained to the interior of the building, which would be exactly sufficient to provide for all losses at 60° — 30°, then the inside temperature would vary in some such manner as is shown in the upper curve.

The actual facts are therefore so exceedingly complicated that it is not surprising that the commercial heating engineer has failed to grasp their very important bearing on the question of the fulfilment of guarantees. He has endeavoured to solve the difficulty in a somewhat feeble manner by laying down a table of what he calls "corresponding" differences of temperature, according to which, if an apparatus will maintain 60° with 30° outside, the same apparatus will maintain 62° with 34° outside, 63° with 36° outside, and so on.

This contention is based on no sound reasoning whatever. From a theoretical standpoint the heating engineer can only claim a reduction of a small fraction in the temperature difference for a high external temperature with a constant radiator temperature, and no reduction at all with a constant heat supply.

This will be understood from the following calculation:—

It is assumed that a radiator of A square feet will maintain a standard 30° to 60° difference in a certain room of exposure factor E. Required the temperature which it will maintain when the outside temperature is t°.

(*a*) With the same radiator temperature T as before.

(*b*) With the same total heat delivery to the interior of the room.

Let x be the required temperature, then

$$A\,K\,(T - x) = E\,(x - t)$$

or

$$A\,K\,T - A\,K\,x = E\,x - E\,t$$

or

$$x = \frac{A\,K\,T + E\,t}{A\,K + E}$$

also

$$A\,K\,(T - 60) = E\,(60 - 30)$$

whence eliminating E, A, and K

$$x = \frac{30\,T + t\,T - 60\,t}{T - 30}$$

Where T is taken as 160°, this gives the following table:—

External temperature.	Internal.
30	60
32	61.5
34	63
36	64.5
38	66
40	67.5
42	69
44	70.5

This calculation shows that from the point of view of strict theory, when the radiator temperature is the same as before and the output of heat therefore reduced, the internal temperature maintained should rise 1.5° for every rise of 2° outside. The heating engineer claims, however, that the figure should be about 1° or less. As a fact, the only justification which the heating engineer has for claiming any reduction at all is the assumption that the radiator temperature is at a maximum, and the fact that the ultimate result can only be obtained by maintaining the new conditions for a prolonged period, and that such a period of prolonged test is impracticable.

When, however, the output of heat from the apparatus is the same as before, the radiator temperature is correspondingly higher, and the temperature difference maintained should be exactly the same for a high as for a low temperature.

It will thus be seen that the entire theory on which the heating engineer is working is unsound and wholly unsatisfactory. It corresponds to no fact of science, and the sooner he decides to establish his guarantees on a sound scientific basis the better it will be for himself.

Testing of a Heating Apparatus.

A consideration of the above facts will show clearly that it is in the highest degree futile to attempt in a relatively short time to make a test of the adequacy of a heating installation to fulfil such conditions as are usually specified by starting the apparatus in a cold building and measuring the temperature attained after a run of a few hours or even of a few days. The calculations are made on assumed conditions, and if those conditions are not approximately fulfilled it is not only useless, but even perfectly futile to "test" whether the calculated result is obtained.

No adequate practical test on these lines can be made, even with a constant outside thermometer, unless the apparatus has been worked continuously at full power for a fortnight before the actual readings are taken. Even so the only conditions in which an accurate result can be secured are that the outer temperature during the whole or the greater part of this period should be near to that which prevails when the actual readings are taken. If this is not done the heat in the walls produces results which no one can gauge.

But even this does not express adequately the whole difficulty. A consideration of the above diagram will show that in no circumstances can the value of the internal temperature at one particular time, or even during the whole of one day, form any guide whatever to the power of the heating apparatus apart from the previous heat history of the walls.

The only conditions which will permit a test to be taken which will enable the adequacy of a heating apparatus to maintain a given temperature condition in a building would be that observations of internal and external temperature should be kept, with the apparatus working at full power continuously day and night for three weeks or one month, such readings being taken after the apparatus had been at work for a fortnight at least. As a practical proposition the mere suggestion of a test of this duration is of course absurd.

An analysis of these fundamental facts shows that if the temperature observations are taken when the outside temperature is rising the internal readings will show a result very much less than they should show. If, on the other hand, the internal readings are taken when the outside thermometer is falling the result may be very much more favourable than the power of the apparatus would justify. They therefore show that neither the one nor the other are of any value whatever as an indication of the power of the apparatus unless they are combined with a very exhaustive analysis of the heat history of the walls.

It would be easily possible to make a test on an apparatus with a rapidly rising outside thermometer which would show the internal temperature actually lower than the external temperature, even although the apparatus might be in fact fully equal to its required duty. Thus, if in Fig. 100, instead of suddenly rising 20° at the point A, the external thermometer had actually risen 30°, and this is not by any means an unknown variation between two successive days in this climate, it is evident that in these conditions the external temperature would have been actually higher than the internal temperature, although by assumption the apparatus is of the theoretically correct power, and although it has been by assumption continually at work at full power during the previous month.

That these test conditions cannot commonly be secured does not in the least alter the fundamental fact that unless they are fulfilled no test can give any proof or disproof of the adequacy or otherwise of the installation to fulfil the required conditions. The temperature result actually obtained, instead of being an indication of the power of the heating installation is merely a sort of summarised history of the heating of the walls during the previous fortnight.

This is one of the most important facts connected with the heating of a building, and it is surprising that it is not more generally recognised.

Conditions of Contract.

Let us now consider, in the light of these considerations, the method commonly adopted for specifying the contract power of an apparatus in current practice. This method is that the apparatus shall be of such power that it will maintain a certain internal temperature in the building for which it is designed, in certain incompletely specified external conditions.

It is probable that the confusion of thought engendered by the essential defects of this method of specifying in practice the power of an apparatus is more responsible for hindering the development of the knowledge of heating than any other single cause. The essentially vicious feature from an intellectual point of view is that it fixes the attention of the engineer on the internal temperature as a criterion of the power of his apparatus, or perhaps on the difference between the external and internal temperature.

It has been shown that this is a variable and capricious figure, from the value of which at any particular time no deductions can be drawn as regards the power of the apparatus to fulfil given conditions.

It combines into one, two fundamentally distinct, unconnected and independently varying functions, the power of the heating apparatus and the history of the conditions surrounding the building. It is not therefore possible, according to a well-known principle of science, that such a reading can give any indication of the value either of the one or of the other, or indeed that any accurate deduction can be drawn from such a figure in any circumstances whatever.

Either of the two alone is more than sufficiently complicated and difficult to estimate when standing alone, but when taken in combination the complication becomes so great that it is futile even to make any attempt to analyse the meaning of any result that may be secured.

The universal adoption of this absurd criterion is unquestionably as much against the interests of an honest client as of an engineer. As far as the client is concerned it prevents him from knowing what he is buying, or what his outlay is going to cost him in upkeep, he cannot tell what value he is obtaining for his money or whether the contract has been properly carried out or not. He has generally to pay from 25% to 50% more for his requirements than he would have if a more reasonable method were in use. For the existence of this criterion and the possible necessity for demonstrating what cannot be demonstrated, compels the heating engineer to put into a client's house an apparatus from 25% to 50% more powerful than is necessary for ordinary requirements.

As for the heating engineer, this criterion of the power of an apparatus is at the root of the greater part of the business troubles with which he is afflicted. It is impossible for him to demonstrate the performance of his undertaking unless he supplies an installation far more powerful than is commonly necessary or desirable. It puts it into the power of any unscrupulous client to delay or refuse payments justly due on work completed, which leads to innumerable unseemly wrangles between architect, client, and engineer. It is a direct premium on ignorant and inferior work, since it is impossible to test with any certainty whether or not the contract has been performed. It imposes on the engineer the necessity of spending large sums of money annually in making useless calculations, which he ought not to be compelled to make.

The amount of heat which will be lost from any building in any given and exactly defined conditions as to temperature, humidity, air movement, and the like, is a quantity which is fixed by the immutable laws of nature. It is a function which depends solely on the quality of the building. The conductivity of the material of which it is constructed, the degree of leakage of air, the influence of the external weather and all surrounding objects, in directing or deflecting air currents, and the multifarious other conditions surrounding a building, some of which have been alluded to in the earlier parts of this work, have all a real and positive effect on the amount of heat lost, to an extent which it is impossible to calculate with certainty.

It can only be determined exactly by direct experiment on the building itself in the given conditions. Such an investigation is of the most difficult description.

The quantity of heat to be supplied should not be determined by the heating engineer at all, but by expert professional advice on the responsibility of the owner of the building. The matter has nothing to do with the contractor as such.

This is the system adopted in America, where the owner of an important building invariably secures advice on such matters from an expert and not from a heating contractor.

The responsibility of the heating engineer should be confined to delivering to each room heated a scheduled number of B.T.U., which is a quantity which can readily be demonstrated from the temperature of the flow and return pipes and the size of the radiator or coil. There his responsibility should end.

If the client wishes to have more heat he can have it by paying for it.

The present scheme operates so as to throw the entire responsibility, professional and otherwise, on the shoulders of the engineer, who is often hardly capable of sustaining that responsibility. The practice throws all the advantages into the hands of an unscrupulous client or architect, who can obtain his expert advice for nothing, by inviting a number of good firms to compete for the work without undertaking any responsibility himself. He can obtain the advice of a good firm, which he can hand on to an inferior and cheaper firm, and escape from or delay payment when he so desires.

Instead of a heating contract becoming a plain business contract, as it should be, it becomes a speculative bargain on the part of the engineer. As a speculative bargain it may be justifiable enough. If he fails to put in an apparatus of sufficient power, he can always rectify the mistake, even though it may be no fault of his own, at a certain cost to himself, and he is able in that event to carry out the contract at a loss.

He may cover his average liability by the addition of a certain percentage of his cost, which his experience should enable him to assess.

The essentially vicious feature of the system is that except in very special cases it prevents the engineer from demonstrating his performance of his contract, unless he has in fact largely exceeded it. It invites an unscrupulous client to refuse payment for the apparatus pending the arrival of cold weather, and gives rise to all sorts of disputes as to open windows, period of heating, interchange of air, draughts, variations, and the like.

It is, however, always *possible* for the contractor actually to carry out his contract if he fails to do so in the first place.

Guarantees of Fuel Consumption.

When, however, the contractor guarantees that not only will he maintain the required temperature in the room under the specified conditions, but will do so *at a given maximum fuel consumption,* he is undertaking a liability of quite another order. It is one which may be physically impossible for him to carry out in any circumstances whatever.

For instance, the application of the standard co-efficients shows that a building should lose, say, 1,000,000 thermal units per hour, and if the engineer calculates his fuel consumption accordingly, it is quite conceivable that owing to the inaccuracy of the co-efficients and of the estimated interchange of air, and other causes, the building would actually lose 30% more than this, say, 1,300,000 B.T.U. This is by no means an unknown variation. It would then be a physical impossibility for the engineer to maintain the given temperature with the guaranteed fuel consumption. No expenditure of money would enable it to be done. His legal liability in damages for failing to carry out his guarantee might easily, and generally would, largely exceed the entire cost of the installation.

It would be represented by a determined and unscrupulous client as a capitalised value of the necessarily increased expenditure in fuel. The legal assessors cannot be expected to appreciate the scientific aspects of the matter, and there are always, unfortunately, heating engineers to be found sufficiently unscrupulous or ignorant, or both, to help such persons to substantiate claims of this character.

It is as if a civil engineer should contract to excavate a hole in the ground 35 feet long by 20 feet wide by 10 feet deep, and should guarantee not to remove more than, say, 400 tons of clay in forming the hole. The weight of material which he would be *compelled* to remove in order to fulfil his contract would depend exclusively on the density of the clay. The efficiency of the work of the contractor would have nothing whatever to do with it. He has no more control over the amount of material that he may be compelled to remove in order to carry out his contract than the heating engineer has over the amount of heat he may be compelled to supply to a building in order to maintain it in a certain condition as to temperature.

The "guarantee" is in reality a mere reckless speculation in the one case of what the density of the clay may prove to be, and in the other of what the combined heat-dissipating properties of the walls and weather conditions may be.

It is a liability which all heating engineers should definitely refuse to undertake, especially having regard to the essential uncertainty of the quantities of heat which may be involved, and to the dangerous legal position in which such a guarantee may involve them.

CHAPTER XVIII.

COMBUSTION OF FUEL.

Quantity of Heat from Fuels.

PRACTICALLY all the heat used in heating buildings, or for any other purposes, is derived from the combustion of fuel, chiefly, of course, coal and coke. It is therefore important to consider carefully the phenomena of combustion and the total amount of heat that can be obtained from the combustion of a given weight of fuel of a given kind, and the proportion of this total heat which can be utilised under different conditions.

Heat is obtainable from fuel owing to the fact that it contains latent chemical energy, which is liberated in the form of heat on the combination of the combustible elements of the fuel with the oxygen of the air. The amount of latent chemical energy existing in one pound of the various essential elements of which fuel is composed will be seen from the table, which is based on very careful and exhaustive experiments. These quantities of heat represent the total obtainable by means of a perfect calorimeter when all the energy of the material obtainable by combination with oxygen is converted into heat.

Table of Heating Powers of Elements in Combination with Oxygen.

Name of Substance.	Weight of pure Oxygen per pound of combustible.	Weight of products of combustion. Pounds.	BTU given off.	Chemical formula of products of combustion.	Volume of 1 pound of products at 32°F. cubic feet.	Specific heat of 1 pound of products.
Hydrogen... ...	8.0	9.0	62,032	H_2O	19.913	.475
Carbon (Coke) ...	2.66	3.66	14,485	CO_2	8.157	.216
Sulphur	1.00	2.00	4,032	SO_2	5.513	.155
Carbon burnt to CO	1.33	2.33	4,452	CO	12.804	.248

Fuels of all kinds consist essentially of carbon, hydrogen, and sulphur in various proportions, together with trifling quantities of some other combustible elements. There are also dead or neutral elements in fuel, such as oxygen, nitrogen, water, and solid incombustible ash. These latter do not take part in the combustion, except that the oxygen already in the fuel may form other combinations during the process. The other neutral substances simply operate to diminish the intensity of the combustion.

The heat given off by the combustion of any given quantity of any fuel depends altogether on the chemical energy of the substances of which the fuel is composed. In so far as any of the constituents of the fuel are already combined with oxygen, they are of no value as heat producers. In so far as the combustible elements in the fuel are themselves combined together that combination in itself detracts from the value of the fuel to some extent.

The combination of any chemical element with *any* other element always results in the giving off or absorption of some heat. If a fuel consists of uncombined elements *mixed* together in certain proportions, then the total heat which would be given off by its combustion would, of course, be equal to the joint quantity of heat given off by its constituents separately during their combination with oxygen.

A fuel, however, is not in fact so simple a matter as this, being in the main composed of free carbon and various *compounds* of carbon and hydrogen. The carbon and the hydrogen are themselves already combined in definite chemical compounds in the fuel, and the energy due to that combination is already lost, for it has already been given off. The heat given off by the combination of carbon with hydrogen is not very great, so that the reduction in the value of fuel, owing to this cause, is not serious.

When the fuel is combined with oxygen by being burnt, the heat breaks up the previously existing combinations between carbon and hydrogen, since they are not such strong combinations as combinations of carbon and oxygen on the one hand, and hydrogen and oxygen on the other. The total heat actually given off is in reality

The heat due to the combination of the given quantity of carbon with the corresponding quantity of oxygen + the heat given off by the actual quantity of hydrogen with its corresponding quantity of oxygen — the very small amount of heat which has already been lost by the carbon and hydrogen owing to their combination with one another.

As a matter of fact, however, it is not possible to take account of this small amount of energy lost, owing to previously existing combinations, and all fuels are calculated as though they consisted of simple mixtures of carbon and hydrogen.

When oxygen is present in the composition of a fuel it is always found in combination with hydrogen. It thus neutralises a portion of the most valuable constituent of the fuel for heat producing purposes. If it is known that the oxygen exists in the fuel as water, which is usual, the weight of the water is to be subtracted from the weight of the fuel before the calculations are made. If the percentage of oxygen only is given it must be assumed that it is combined with hydrogen as water, and the necessary quantity of hydrogen is calculated and subtracted from the total hydrogen in the fuel in order to arrive at the available hydrogen.

The table shows that in the combustion of hydrogen one pound of hydrogen by weight combines with eight pounds of oxygen by weight to form nine pounds of steam by weight, and the process of combination gives off about 62,000 B.T.U. of heat. Similarly, one pound of carbon combines with 2.66 pounds of oxygen, forms 3.66 pounds of carbonic acid gas, and during the process gives off heat of about 14,500 B.T.U.

If carbon is not fully burnt, that is, if there is not sufficient air supplied to the carbon to consume it completely, under certain conditions it is converted into "carbonic oxide," and gives off about one third of the amount of heat given off when it is completely burnt to carbonic acid.

Properties of Fuels.

British *coal* generally contains from 70 to 85% of carbon, from 3 to 7% of hydrogen, 1 to 2% of nitrogen, 1 to 5% of sulphur, 1 to 20% of oxygen, and from 2¼ to 15% of ash.

The best Welsh coals contain as much as 90% of carbon, and generally 5% of hydrogen, 1½% of ash, 2% of nitrogen, 1% of sulphur.

Coke.—The best coke made from bituminous coal is crystalline and highly porous, having a bright greyish colour with metallic lustre, and is very hard. Purest coke consists almost entirely of carbon, having a minute percentage of sulphur and ash in the proportion of about 97½% of carbon, 1% of sulphur, and 1½% of ash. The average of a large number of samples of coke shows 93.5% of carbon, 1.2% of sulphur, and 5.3% of ash. Coke is capable of absorbing up to 20% of its own weight of water, without being exposed directly to rain. When exposed to rain it may absorb as much as 50% of its weight in water, the greater part of which may afterwards be drained off or evaporated, leaving about 5 to 10% in the coke. The value of coke as a fuel depends very largely on the percentage of carbon which it contains.

Wood is somewhat unsatisfactory as a fuel, on account of the large volume of flaky ash which it gives off, and the tendency to difficult combustion in consequence, also on account of the large amount of water which it can contain. When it is thoroughly desiccated the temperatures which can be obtained from it are considerably increased. By exposure to temperatures up to 300° F. the wood is desiccated. When heated to over 300° F. the gaseous elements are given off. At 650° F. the wood produces charcoal, black and brittle. All the gases, however, are not completely given off until the temperature is still further raised.

Peat is rarely used as a fuel, and cannot be used until it is thoroughly dry. In its natural condition 78% of its entire weight is water.

Petroleum is a hydro-carbon liquid of a specific gravity of about 0.85. It is composed of a large number of different hydro-carbons, the lighter of which are frequently distilled off, leaving a residue which is sometimes used for heating purposes, not much, however, in this country. It is, perhaps, the most powerful of all fuels in point of B.T.U. per pound.

Coal gas varies considerably in composition. It is considerably used for local heating, and for cooking as well as lighting.

The average composition is shown in the table in the Appendix, as well as the heating power. The latter varies considerably, according to the source of the gas and treatment and manufacture.

Mechanical Properties of Fuel.

The only fuels with which the heating engineer has commonly to do are coal, coke, and sometimes wood, peat, and petroleum.

A few remarks on the qualities of these various fuels may therefore be desirable.

Coals may generally be divided into five classes. (1) Anthracite, which consists largely of free carbon; (2) dry bituminous coal, having from 70 to 80% of carbon; (3) bituminous caking coal, having 50 to 60% of carbon; (4) cannel coal, chiefly used for gas making; and (5) brown coal. The latter is not much used in this country.

Anthracite is the most valuable form of coal for heat producing purposes. It is supplied generally in small lumps, though frequently in large pieces, and has a bright brittle appearance. When rapidly heated the lumps fall to pieces, though if slowly heated they can be heated up to the temperature required for ignition without changing their form. They yield little flame of a bluish colour, and an intense local heat in consequence. The fuel is generally difficult to ignite.

Dry bituminous coals contain a relatively small proportion of volatilisable matter, about 15%, and are easily ignited. The lumps of coal swell and blister when heated. Their combustion is rapid.

Bituminous caking coals contain a large proportion of volatilisable matter, and burn with a long smoky flame. Under heat they swell considerably, and give off strong pieces of coke and ashes. They are not much used for heating purposes, but chiefly for household purposes and gas making.

Coke, which is often the most suitable for heating engineers' purposes, has very variable properties, according to the coal from which it is derived.

When coal is thrown into a boiler furnace its behaviour varies according to its quality.

With anthracite coal it disintegrates very slowly, unless it is very rapidly heated, when it is apt to split into small pieces. There is very little volatile matter to distil and therefore the coal burns slowly.

With the dry bituminous coals, disintegration and distillation take place somewhat faster, and with the softer varieties of coal almost instantaneously. The volatilisable constituents often form tar and smoke and soot, which condense or cling to the heat receiving surface and prevent the passage of heat.

These differences constitute the reason why anthracite coal is so much more valuable for heating purposes than coal of a bituminous character. In the case of anthracite the combustion is complete almost before the gases have left the fuel, whereas with bituminous fuels the flame completes the process, and the flame is apt to come into contact with the cold surface and be chilled before the combustion is quite complete, resulting in the formation of smoke and soot.

Calculation of the Heating Power of a Fuel.

The heat given off by the combustion of one pound of any ordinary fuel depends therefore on its composition. In order to calculate the heat obtained from one pound of such fuel, having a given percentage composition, it is sufficiently near for practical purposes to determine the actual weight of carbon, free hydrogen, and other constituents in one pound, and calculate the heat which would be given off if each were burnt separately.

The essential combustible elements in fuel are carbon, hydrogen, and sulphur. The percentage of sulphur is in all cases very small, and for approximate calculations may be neglected. We are left, therefore, with the carbon and free hydrogen.

If the percentage composition of the fuel is given as follows, oxygen O%, carbon C%, hydrogen H%, and incombustible ash (with which may be reckoned the sulphur), A%, it is clear that the actual weight of carbon in one pound of the fuel will be the fraction $C/100$ of one pound. Similarly of hydrogen, the weight will be $H/100$ of one pound. The weight of oxygen will be $O/100$, with which will be combined ⅛ of its weight of hydrogen, or $O/800$.

Hence the free hydrogen will be

$$\frac{H}{100} - \frac{O}{800}$$

Hence the total heat evolved will be

$$145\ C + 620\ (H - \frac{O}{8})$$

The heat value of different fuels is therefore roughly given by Fig. 102 in which the abscissæ are percentages of *free* hydrogen, and the ordinates the total B.T.U. in one pound of fuel, a separate curve being drawn for each percentage of incombustible ash, in which is reckoned the

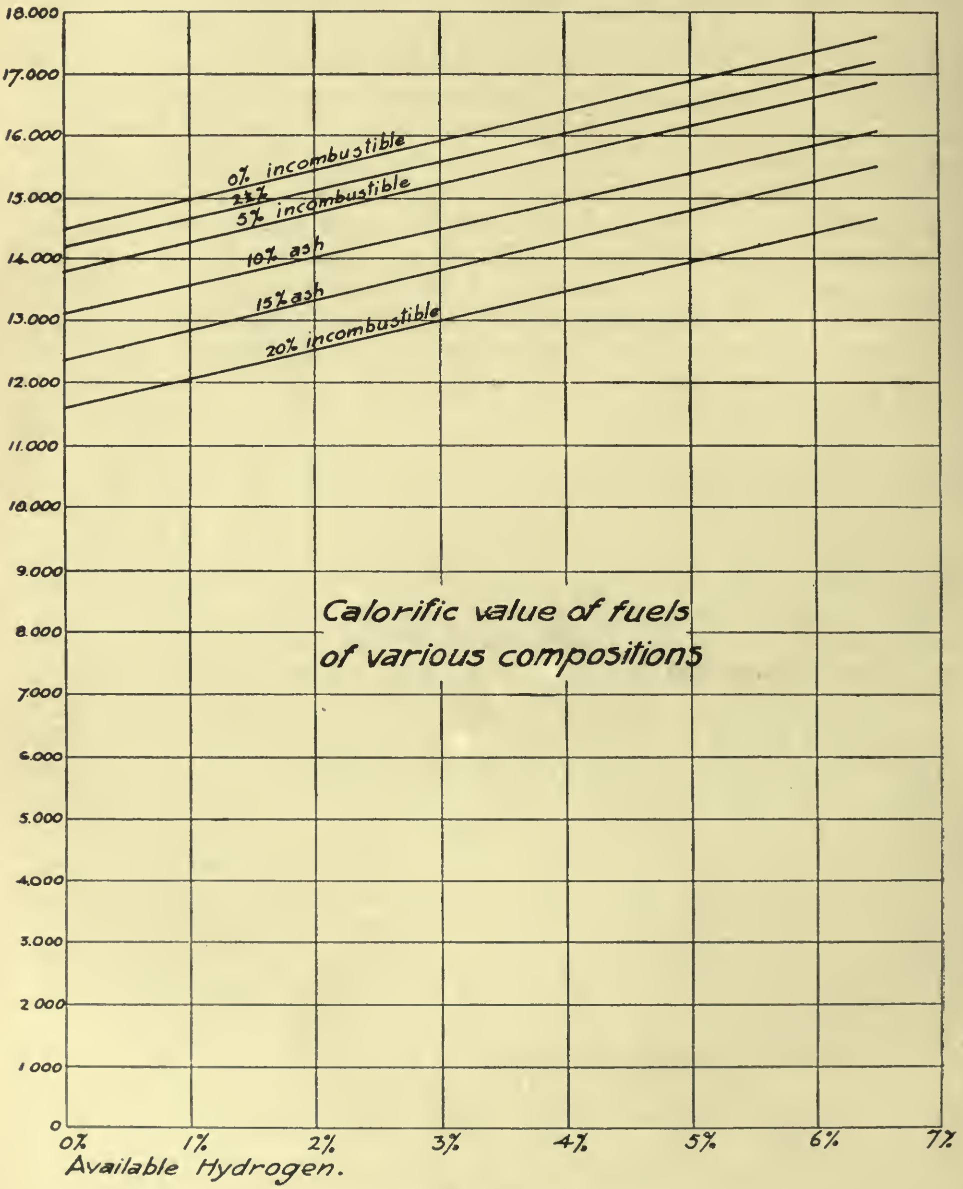


FIG. 102.

sulphur and the water or fixed hydrogen. All the remainder of the fuel is supposed to be carbon.

Table XXIV. gives the approximate composition of different qualities of coal, from which it will be seen that the hydrogen is practically always below 7% of the total.

Temperatures of Combustion.

When any fuel is burnt the total heat generated by the combustion is in fact given off partly as radiation and is partly communicated to the products of combustion, raising their temperature to a high point.

If the fuel is burnt in the open air a considerable proportion of the heat escapes as radiation. When the fuel is burnt under such condition that radiant heat cannot escape in any other way, the whole of the heat generated is presumed to be absorbed in raising the temperature of the gases. This is the case when heat is given off in a furnace. It is probable that in fact a large part of the heat given off by the glowing fuel at the top of the fire reaches the furnace walls without raising the temperature of the products of combustion.

The interchange of radiation with the relatively cold walls of the furnace causes a considerable reduction of temperature at the surface, but the radiant heat given off in the body of the fire is at once absorbed by the rest of the burning fuel and is radiated backwards and forwards until it is taken up by the furnace gases passing through the incandescent fuel and carried up above the top surface of the fire.

It should be clearly understood that the temperature indicated by a pyrometer in the furnace is merely that of the instrument itself, subject as it is to the varying influences of hot gases, hot and cold radiation from the glowing fuel and the cold walls respectively. The only way to ascertain the actual temperature of the furnace gases would be to draw off a measured volume of them and abstract the heat they contain in a calorimeter. It is probable that from one quarter to one half of the total heat generated in a furnace passes to the walls by radiation.

The calculations of the "theoretical temperatures" produced in a furnace are made on the assumption that the *whole* of the heat generated by the combustion of the fuel is absorbed by the gaseous products, which, as explained above, is not strictly true. It is impossible, however, in practice to distinguish between the heat given off in radiation and by convection.

If H is the heat given off in the combustion of one pound of the fuel,
S the mean specific heat of the products of combustion,
W their weight,
t_1 the temperature of the air before passing through the furnace, and
t_2 the temperature of the gases afterwards.

Since the specific heat of the gases does not differ considerably from the specific heat of air, the approximate equation will be as follows :—

$$W (t_2 - t_1) S = H$$

neglecting relatively unimportant considerations for simplicity.

Consider the theoretical temperatures obtained on the assumptions noted above, by the combustion in pure oxygen of 1 lb. of carbon and hydrogen.

One pound of carbon requires for complete combustion 2.66 pounds of oxygen, producing 3.66 pounds of CO_2.

The specific heat of CO_2 is 0.216 and the B.T.U. given off from the complete combustion is 14,500.

So we have

$$14{,}500 = 3.66 \times 0.216 \times (t - 60^\circ)$$

$$(t - 60) = \frac{14{,}500}{3.66 \times 0.216} = 18{,}350^\circ$$

One pound of hydrogen requires 8 lbs. oxygen.

B.T.U. produced 62,000.

Specific heat of water vapour 0.475.

Weight of vapour produced = 9 lbs.

Hence temperature produced

$$(t - 60) = \frac{62{,}000}{9.0 \times 0.475} = 14{,}500^\circ$$

The above figures give the theoretical temperature which could be obtained by the combustion of a given quantity of fuel with pure oxygen. Such temperatures are, of course, never obtained in practice.

When these elements are burnt in air a large reduction of this temperature is caused by the fact that the gas supporting the combustion is not pure oxygen, but a mixture of gases, of which only 1/5th part by volume consists of oxygen. This at once reduces the theoretically attainable temperatures of combustion very greatly.

A large part of the heat evolved by the combustion is taken up in the heating of a large mass of inert gas, especially nitrogen, of which air is largely composed. In any case, therefore, even when fuel is burnt under the best conditions by the minimum quantity of ordinary air, the temperature attained must be very much lower than would be the case with pure oxygen.

There are other causes which still further limit, in practice, the temperature which can be attained by the combustion of fuel with air.

It is not generally possible to burn solid fuel *completely* unless the quantity of air supplied to it is considerably in excess of the chemically necessary minimum. In practice fuel is almost always in the form of lumps of coal or coke, and it is burnt on a grate in a heap. For practical convenience it is necessary to have a considerable quantity of fuel on the furnace at one time, in order to avoid having to stoke at frequent intervals.

Now when air passes through a thick layer of incandescent fuel some of the carbon-di-oxide which is produced in the lower sections of the fire remains in contact with the incandescent fuel above it at a high temperature. It is a chemical property of carbon-di-oxide that in these circumstances it will take up a further quantity of carbon from the glowing fuel, and it carries it off only partly consumed in the form of the gas carbon-mon-oxide.

If arrangements are not made for burning it, this partly consumed carbon is carried into the atmosphere and wasted.

The actual number of thermal units given off in converting one pound of carbon into carbon-mon-oxide is only about 4,460 B.T.U. as against 14,500 B.T.U. evolved by completely burning the carbon into carbon-di-oxide, representing the complete saturation of the carbon with oxygen.

When fuels are used which consist in any degree of volatile combustible constituents, these are at once distilled when the fuel is subjected to the heat of the furnace. In order that they shall be completely burnt, or oxidised, it is obviously necessary that at some period during its passage through the furnace, *every* molecule of the volatile substance shall come into contact with a suitable number of molecules of oxygen. If this effect is to be secured with the theoretically minimum quantity of oxygen, it involves an absolutely perfect mixture or interdiffusion of the two gases. But as the air must be introduced in the form of a number of streams, stratification of the mixture of air and combustible gas is produced, and the interdiffusion has not time to become complete before the mixture has passed into a region which is too cold to maintain combustion.

In order to avoid waste from all these causes and to utilise the carbon to the full, it is therefore necessary in practice that there should be an *excess* of air passing through the incandescent fuel over and above such an amount as would contain only just enough oxygen to consume it completely. The amount of air found in practice to be necessary to secure complete combustion is from 1.5 to 2 times the theoretical quantity. There is, therefore, always in the waste furnace gases a considerable quantity of uncombined oxygen, and also a large quantity of nitrogen, so that the temperatures which might be obtained under perfect conditions, if such were possible of attainment, are very largely reduced in practice.

Other kinds of fuel—liquid and gaseous—can however be used in furnaces in such a way that very little more than the theoretical quantity of air will suffice to burn them completely, because liquid or gaseous fuel can be so subdivided as to be brought into more intimate contact with air than is possible in the case of solid fuel.

Therefore the temperatures which can be attained by the use of liquid and gaseous fuels are much greater than with solid fuels. But liquid and gaseous fuels are at present so much more expensive than solid fuels that they cannot be generally employed.

Temperatures Practically Attainable.

The theoretical temperature of combustion, when air is used instead of pure oxygen, may be ascertained as follows :—

Since one pound of carbon requires for complete combustion 2.66 pounds of oxygen, and since air is composed by weight of 24 parts of oxygen to 76 parts of nitrogen, the theoretical quantity of air required to consume one pound of carbon is

$$2.66 \times \frac{100}{24} = 11.1 \text{ pounds of air,}$$

which represents

$$11.1 \times 13.1 = 145 \text{ cubic feet at } 60^\circ \text{ F.}$$

The amount of heat evolved in the process of combustion is as before, 14.500 B.T.U. The products consist of 3.66 pounds of CO_2, and 8.44 pounds of nitrogen, a total of 12.1 lbs. The specific heat of CO_2 is 0.2164, and of nitrogen 0.244, therefore the specific heat of the products of combustion is

$$\frac{3.66 \times 0.2164 + 8.44 \times 0.244}{12.1} = .236$$

The temperature attained will therefore be found as follows :—

$$14{,}500 = 0.236 \times 12.1 \times (t - 60^\circ)$$

$$t - 60 = \frac{14.500}{0.236 \times 12.1} = 5{,}060^\circ \text{ F.}$$

A similar calculation for hydrogen is as follows :—

1 lb. H requires 8 pounds of oxygen, or

$$8 \times \frac{100}{24} = 33.3 \text{ lb. of air.}$$

The products are 9 lbs. water vapour and 25.3 lbs. nitrogen. Specific heat of water vapour = 0.475, of nitrogen 0.244, of mixture

$$\frac{9 \times 0.475 + 25.3 \times 0.244}{34.4} = \frac{4.275 + 6.17}{34.4} = \frac{10.445}{34.4} = 0.305$$

$$t - 60 = \frac{62{,}000}{0.305 \times 343} = 5{,}910^\circ \text{ F.}$$

Assume a sample of coal is burnt, consisting, say, of 85 parts of carbon and 5 parts of hydrogen, and 10 parts by weight of neutral components, ashes, etc.

Then the theoretical temperature of combustion is found as follows :—

Total heat emitted when 0.85 pounds of carbon are fully combined with 0.85 × 2.66 lbs. = 2.26 lbs. oxygen, or

$$2.26 \times \frac{100}{24} = 9.4 \text{ lbs. air}$$

is

$$14{,}500 \times 0.85 = 12{,}300 \text{ B.T.U.}$$

Total heat emitted when 0.05 pounds of hydrogen is fully burnt with 0.05 × 8 = 0.4 pounds of oxygen, or

$$0.4 \times \frac{100}{24} = 1.67 \text{ lbs. air}$$

$$= 0.05 \times 62{,}000 = 3{,}100 \text{ B.T.U.}$$

Total heat from 1 lb. coal = 15.400 B.T.U.

The theoretical temperature of combustion with a minimum of air is therefore

$$t - 60 = \frac{15{,}400}{0.246 \times 11.97} = 5{,}250^\circ \text{ F.}$$

The minimum quantity of air for combustion of ordinary coal, consisting of 85 parts of carbon, 5 parts of hydrogen, and 10 parts of incombustible material, requires, as has been shown, 9.4 + 1.67 = 11.1 lbs. of air to effect its complete combustion.

The volume of this weight of air at any temperature can be found from the equations given in the chapter on "Properties of Air."

It is found to be necessary in order to secure complete combustion in practice, that the quantity of air supplied to the fuel should be from 1.5 to 2 times the theoretical minimum. The temperature attained can be calculated in the following manner :—

Consider the combustion of the same sample of coal as before, with 1.5 times the minimum air supply. Here, as before,

Total B.T.U. emitted from 1 lb. coal, 15,400 B.T.U.

Total weight of products of combustion, 1 + 11.1 × 1.5 = 17.6 lbs.

Specific heat of products, approximately, 0.246.

$$\text{Temperature attained, } \frac{15{,}400}{0.246 \times 17.6} = 3{,}550^\circ \text{ F.}$$

Similarly with twice the necessary minimum

$$\text{Temperature attained} = \frac{15{,}400}{0.246 \times 23.2} = 2{,}700^\circ \text{ F.}$$

In an ordinary heating apparatus the products of combustion are rejected at a temperature of from 300° F. to 500° F., sometimes up to

600° F. The minimum volume of the products of combustion will therefore be approximately, at 600° F.

$$\frac{11.1}{0.0863} \times \frac{1,061}{461} = 287 \text{ cubic feet,}$$

or calculated at 0° F., 128 cubic feet
or calculated at 60° F., 145 cubic feet.

For general practice it is sufficient to reckon that 300 cubic feet of air at 60° are required for the perfect combustion of one pound of fuel, that is double the theoretical minimum, and the chimney ought to be designed capable of passing this quantity of air at the greatest temperature at which the products of combustion ever reach, which may be taken to be 600° F.

If the products of combustion are above this temperature it is evidence of wasteful management.

When a boiler is being forced the products of combustion ought not to exceed 550° F., and for ordinary working they ought not to exceed 300° F., for economical working (see Fig. 103).

Proportion of Heat Utilisable.

The above particulars show the total quantities of heat obtainable from one pound of combustible, and the highest temperatures which can be theoretically obtained by their combustion in oxygen and with a minimum quantity of air.

Now the proportion of this total heat which could in any case be utilised for the purposes of the heating engineer is considerably less than the total theoretical quantity of heat. The heat is in practice generally utilised by transferring it to a medium which is practically always either water or air.

When these hot gases are brought into contact with heat-absorbing surfaces at a lower temperature they part with some of their heat to those surfaces.

These products of combustion will part with their heat so long as they themselves are at a higher temperature than the heat-absorbing surfaces. The amount of heat so transferred through a given surface in a given time will be roughly proportional to the difference of temperature between the hot gases and the cold surface.

In order, therefore, to abstract as much of this heat from the products of combustion as rapidly as possible, it is desirable that the temperature of the surface should be as low, the temperature of the gases as high, and therefore the difference of temperature as great as possible.

Assume, for instance, that the heat is to be conducted from the products of combustion to the air of a room at 60° through a thin metal plate of given surface (say, for purposes of illustration, 30 square feet) and that the quantity of heat to be communicated per hour is, say, 20,000 B.T.U., and that all the gases in contact with the plate are so mixed up that they are at a uniform temperature. Then the temperature of the products of combustion will be reduced to such a point that they will communicate 20,000 B.T.U. per hour to the heating surface, which is itself conducting the same amount of heat to the air of the room. In these circumstances the two surfaces of the plate will, after a time, each take up a stationary mean temperature, such that the same number of thermal units will be communicated to the air of the apartment per hour by the outer surface as is conducted through the substance of the plate, and the same as the hot flue gases communicate in the same time to the inner surface at that temperature. (See chapter on "Transmission of Heat.")

Assume that the mean temperature of the heating surface is $t_1°$, the temperature of the air being 60°, and assume that K_1 is the co-efficient of transmission between the heating surface and the cool air.

Then approximately

$$20{,}000 = K_1 \, (t_1° - 60°) \, A.$$

Let t_2 be the temperature of the gases, and K_2 the corresponding co-efficient, then also

$$20{,}000 = K_2 \, (t_2 - t_1) \, A.$$

Then we have, assuming the same temperature on the two surfaces of the plate (the actual difference will in fact be relatively unimportant)

$$t_1 = \frac{K_2 \, t_2 + K_1 \, 60°}{K_1 + K_2}$$

and

$$t_2 = \frac{K_1 \, (t_1 - 60°) + K_2 \, t_1}{K_2}$$

This temperature, t_2, is the lowest point to which these gases can be reduced in the circumstances. Any heat which is left in the gases after they are reduced to this temperature must of necessity be thrown away with the flue gases, unless more heating surface is supplied, whereby the mean temperature of the heating surface is reduced. There is, of course, a practical limit to the amount of heating surface which can be supplied.

These conditions therefore represent the extreme degree to which the total B.T.U. can be utilised in this manner in an apparatus of the above described proportions.

It is clear, therefore, that in order to facilitate the conduction of heat to some other medium, when the heating surface is limited, one condition is that the temperature of the hot gases must be retained as high as possible, for the rate at which the conduction will take place is roughly proportional to the whole difference in temperature between the hot gases and the medium. Another condition is that t_1 should be as low as possible and K_2 as high as possible. In other words, we shall increase the total amount of this heat which can be utilised by increasing the conductivity between the hot gases and the heat receiving surface.

The best way of doing this is to bring the temperature of the heat receiving surface down, so that there may be as great a difference as possible in the temperature between the hot gases and the heat receiving

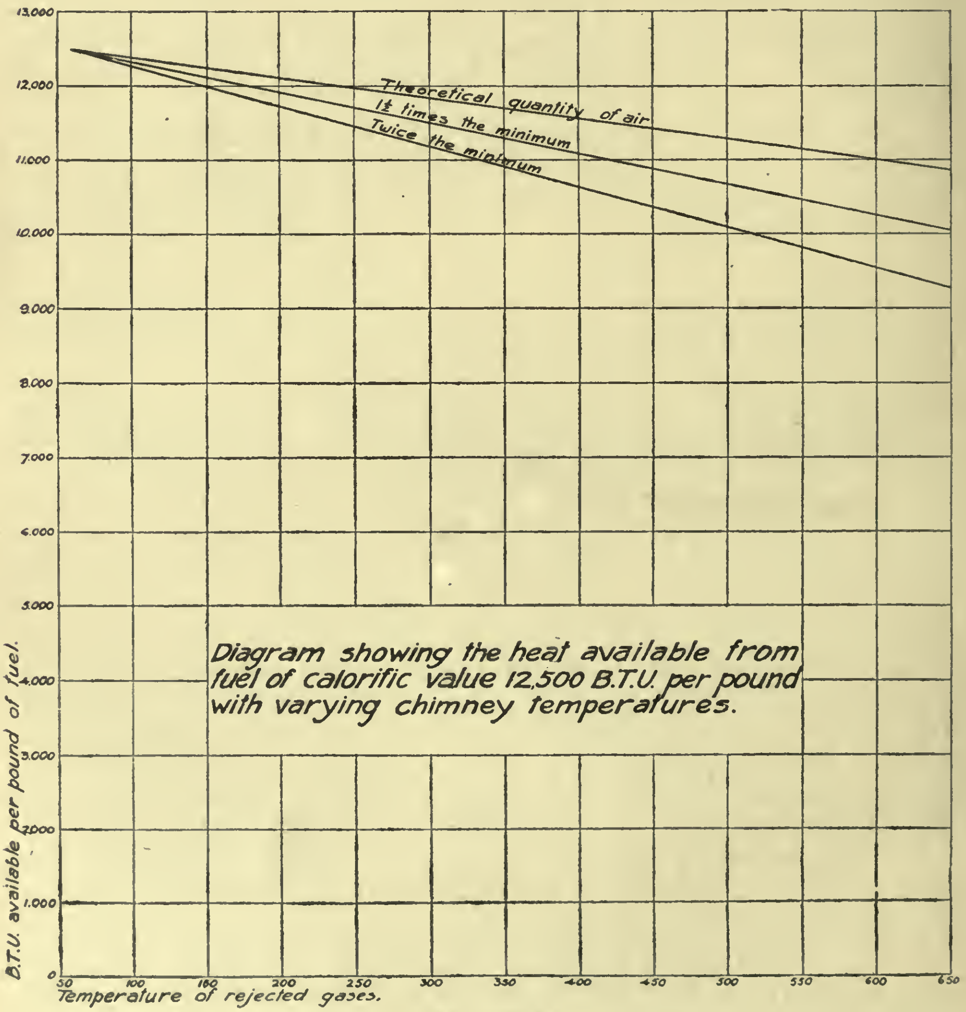


FIG. 103.

surface, and to increase the value of K_1. The value of K_1 depends on the nature of the medium which is abstracting heat from the surface.

With water as the heat receiving medium the rate of conductivity from the surface to the medium is much greater than with air. If the temperature of the water is kept as low and the surface as large as possible, the amount of heat abstracted from the gases will be still further increased.

Fig. 103 shows the variation of the available heat when the temperature of the flue gases varies.

To summarise, the conditions under which the maximum proportion of the total heat generated can in this manner be made available for use, are—

(1) To work with a medium which readily absorbs heat, such as water.

(2) To keep the temperature of this medium as low as possible.

(3) To provide as large surfaces as possible.

(4) To maintain as large a difference of temperature as possible at each point between the hot gases and the plate with which it is in contact, and to make the other conditions of the contact as good as possible.

(5) To increase the conductivity of the plate on each side by providing it with a suitable surface, such that it will readily absorb or part with the heat it contains, including the keeping of the surface free from any films either of soot, tar, deposit from the water, or of hot water or hot air, which would tend to diminish the flow of heat.

These are the points which must be attended to in making any apparatus efficient in utilisation of heat.

If the gases containing the heat are relatively small in volume and at a high temperature, the heat they contain can be more easily and cheaply transferred to another medium than if they are of greater volume and at a lower temperature. The greater their volume and the lower their temperature, the larger must the heating surface be which deals with them. There is a practical limit to the amount of heating surface which can be afforded for any particular duty, and therefore the greater the volume in which the heat is confined the smaller will be the proportion of it which it is practically possible to utilise with a given heating surface.

In order that a sufficiently rapid flow of heat shall pass through the plates bounding the furnace, it is necessary that the temperature of the gases shall be *considerably* higher than that of the medium. The greater the required flow of heat is the greater must this difference be.

It is for this reason that a boiler working at a low temperature is more economical than when worked at a higher.

The method by which air is drawn through the fuel is in almost all cases to maintain the chimney at a high temperature. This produces a suction in the chimney on the principles explained in connection with ventilation.

In order to maintain the draught by this means, the temperature of the gases leaving the heater must be considerable. This method of maintaining the draught necessarily wastes a considerable proportion of the heat of combustion. For these two reasons it is impossible to bring down the temperature of the waste gases as low as is desirable in order to secure a maximum of economy.

Further reasons which tend to limit the percentage of the total B.T.U. contained in a pound of coal or coke which can be utilised in the furnace are that a certain part of all such fuels is apt to be broken into small pieces, owing to the rough treatment it generally receives. These pieces fall between the grate bars and are lost in the ashpit.

Further, the process of combustion tends to block up the combustible materials with cinders and ashes, and in order to remove these it is necessary occasionally to open the fire door and clean the fire, in which process much heat is lost.

Further heat is lost from that part of the surface of the boiler or furnace exposed to air. These matters will be further discussed in the chapter on "Boilers."

For all these reasons it is found in practice that the greatest percentage of the total heat of combustion of the fuel that can be obtained is about 75%, and although certain forms of boiler are on the market which will give a larger percentage of efficiency than this, it is not found in practice to be a practical proposition to obtain a larger percentage in actual work.

The practical conditions therefore necessary to secure the maximum practical result from the combustion of a given weight of fuel, it is necessary to ensure that no more air is passed through the furnace than is absolutely necessary. Further, that the fuel in the furnace shall be so distributed over the grate that it is not possible for any cold air to pass through without coming in contact with incandescent fuel, so as to have some of its oxygen extracted from it and a corresponding quantity of heat generated and communicated to the air passing through the heat-absorbing surfaces.

The heat-absorbing surface should be so large and arranged in such a manner that the temperature of the waste gases leaving the furnace is reduced to the lowest point necessary to maintain the draught in the chimney.

CHAPTER XIX.

ON BOILERS.

A BOILER consists essentially of a combination of a furnace in which fuel can be burnt, with an arrangement of water-cooled plates surrounding the furnace for absorbing the heat generated.

The material of which boilers were formerly made was almost invariably wrought iron (or steel). These materials have been to some extent superseded for heating boilers by the cheaper and more convenient and economical cast iron, though wrought iron and steel are still largely used for this purpose.

Boilers for special purposes are made of copper. The object of using copper is to guard against the corrosive or chemical action of the liquid heated, when this is of such a chemical nature as would affect steel or iron. Copper is also frequently used for the supply of hot water for domestic purposes in order to avoid discoloration of the water by rust, especially where the water used is soft or slightly acidified, as when it is obtained from peaty districts. These are practically the only materials in general use for the construction of boilers of any kind.

The design of a boiler is primarily concerned with two considerations.

(1) How the fuel may be burnt in the best and most efficient manner, so as to secure the highest possible temperature of combustion.

(2) How to arrange the heat-absorbing surfaces so that as much as possible of the total heat generated may be absorbed by the water.

There are, of course, many other practical and constructional problems which the maker has to consider. The shell and all the fittings connected with it must be strong, so that they will not readily get broken. They must be designed for the rough treatment which most boilers receive. The whole apparatus must be so designed that it is convenient to handle, to clean the furnace and the flues, to stoke the fuel on the furnace, to control the draught, and to renew those parts of the boiler which are likely to deteriorate or decay in practical use.

It is a great advantage for a boiler to be constructed in sections so as to be capable of being easily put together on the site, and to be small, so as to occupy as little space as possible. It can then be employed where space and access room, and means for lifting are all limited. For certain purposes, such as the direct heating of hot water supply for baths, lavatories, etc., or for industrial purposes, it is important also that the interior of the boiler should be easy to clean from scale derived from any hard water that is used, which may be deposited on the interior surface of the plates.

For ordinary heating purposes the latter feature is not of great importance, since little or no scale is generally so deposited unless the water is changed. No generally reliable method is known for removing scale from the interior of sectional boilers. Sometimes this may be done by employing boiler composition, but only when the composition of the scale is of a certain quality. If the scale is of lime, of a certain degree of hardness and a certain chemical composition, it cannot be removed by any known means. A weak solution of dilute hydrochloric acid is sometimes useful for removing deposit consisting entirely or mainly of carbonates. If the solution is left in the boiler too long it is liable to corrode the joints and cause leakages.

Other considerations which govern the constructional design of a cast iron boiler, in addition to those above enumerated, are that the various sections of which it is composed shall be easily cast in such a shape as to favour the formation of good castings, and so that the cores can be both readily made and readily removed from the casting after it is made, that the junctions with other sections and other parts of the boiler can be easily and cheaply made, that the various fittings necessary to be added to the assembled sections in order to form a complete boiler, and all parts liable to deteriorate in use shall be strong and conveniently placed and readily renewable when necessary.

In all cases a boiler should be designed with regard to the length of time that a full charge of fuel will last, so that the fire may not need frequent attention. This implies that the furnace capacity must be large enough to hold a quantity of fuel proportional to the desired time between two successive stokings, and to the total quantity of heat to be delivered per hour, the furnace being of such a shape that the fuel once ignited will continue to burn till it is entirely consumed.

The grate bars must be of such design that while retaining all fuel which is in pieces of useful size, they will allow powdered ashes and dust to fall freely into the ash pit.

In the case of wrought iron boilers, the constructional points to be considered by the designer in addition to the foregoing, are that the various plates of which the boiler is composed can be readily welded or rivetted together, and that the designs of the inlets and outlets, inspection doors, and the like, besides being conveniently placed, are such that they can be economically fitted to the boiler.

The Fire Grate and Ash Pit.

The grate is designed with the object of supporting the burning fuel and admitting the necessary volume of air into contact with it, and as far as possible of allowing the burnt ashes to escape from the fire and to fall into the ash pit below.

The design of the grate depends, of course, on the quality of the fuel to be used.

Fuels which are in, or are apt to break into small pieces, should be supported by grate bars with narrower air spaces than in the case of coke or wood, which are generally used in large pieces.

The proportion which the space between the fire bars generally bears to the whole surface of the grate is about as follows :—

For burning wood and turf, 1/5th to 1/7th.

For soft coal, 1/3rd to 1/5th.

For hard coal and coke, 1/2 to 1/3rd of the total grate surface.

The amount of fuel burnt on one square foot of the grate depends on the intensity of the draught and on the character of the grate. If too intense a combustion is maintained on the fire bars, with certain kinds of fuel, it favours the formation of clinker. This clinker is apt to fuse on to the fire bars, especially if any considerable quantity of ash is allowed to collect in the ash pit, and by overheating the fire bars it may destroy them. It is therefore found to be inadvisable in the case of solid fire bars to burn more than about 20 pounds of coal or coke per square foot of fire grate per hour, as an outside figure.

In the case of certain types of boiler, however, this figure may be largely exceeded. The table on page 390 shows examples of the rate of combustion on the fire grates of various kinds of boiler.

If the fire grate is too large there is a danger that certain parts of it may be loaded with a thinner layer of incandescent fuel than the remainder. This causes a loss of efficiency by allowing cold air to come through into the combustion chamber and flues, where it reduces the temperature of the products of combustion, which for economical working should be kept as high as possible.

A large grate tends sometimes to produce a difference in the flow of the air in various parts of the grate. The flow in the centre is apt to be greater than that at the sides, producing a relatively quicker combustion of the fuel at the centre, and hence, after a short time, an irregular distribution of the unconsumed fuel on the grate.

In order to avoid this effect, some engineers think it advisable that the fuel on a large grate should not be uniformly distributed over the whole of the grate, but should be arranged in the form of a shed with the thickest layer of fuel in the centre.

The length of the grate must not be so great that the stoker has difficulty in throwing fuel accurately to the back of it. The greatest length of fire grate which a stoker can generally feed with advantage is about five feet. If the length of the boiler is greater than this losses are likely to be occasioned on this account.

Size of Furnace.

For convenience of operation of a heating boiler, it is necessary that the fuel should not have to be frequently stoked on the furnace. The proportions of the design should be such that a relatively large quantity of fuel can be fed on at one time, the boiler being thereafter left for six or eight hours, and the heat being maintained the whole time.

For instance, a boiler intended to maintain an output of 160,000 B.T.U. per hour for eight hours must have a space actually available for fresh fuel alone (apart from combustion space and igniting bed) of

$$(160{,}000 \times 100/60 \times 8) \div (13{,}000 \times 28) \text{ cubic feet.}$$

The fuel must not be fed into the furnace in such quantity that it fills up the whole of the furnace chamber. There should be a space of 6″ or more above the top of the fuel, or deep enough to permit the perfect formation of flame before it comes in contact with the cold crown of the furnace. If this is not done the flame is chilled before combustion is complete, and the gases escape partly unconsumed.

There must also be a space immediately over the grate of from three to 5 inches deep, which in working will never be free of incandescent fuel, forming a permanent bed of fuel. This fuel is for re-kindling the new charge, and the space it occupies must not be reckoned in calculating the fuel capacity of the boiler.

Smoke and Fume Formation.

It is important from a hygienic point of view that the combustion be effected with as little formation of smoke as possible. It is not consistent with the greatest economy that with ordinary coal fuel there should be *no* smoke whatever, neither is the formation of very heavy smoke consistent with economy. A maximum of economy is obtained by combustion of such a character that a light transparent smoke is formed. Even such smoke as this is, of course, disadvantageous from a hygienic point of view, but from an economical point of view it is better to have smoke of this character than no smoke at all. If there is no smoke whatever it is generally an indication that the excess of air is too great.

With coke fuel, if owing to any abnormal efficiency in the boiler, the temperature of the gases leaving the chimney is reduced too far, they may not be hot enough to secure their dissipation in the air above the chimney top, and they may become so far cooled that owing to their density they fall in a body to the ground, and may produce a good deal of inconvenience.

The production of smoke may often be diminished or entirely prevented by the admission of heated air above the fire. This method, though in certain favourable conditions it may increase the efficiency of the boiler, almost always reduces the total output. If the facilities are misused they may lead to very large waste of fuel. The practice of leaving the fire door partly open is almost always extremely wasteful in fuel.

On Rates of Combustion.

Assume that the total fuel burnt on the grate of the boiler in an hour is W pounds, and the area of the boiler grate being A square feet, $\frac{W}{A}$ is called the rate of combustion on the grate, being the mean number of pounds of fuel burnt per square foot per hour taken over the entire grate. The rate of combustion is actually considerably greater in the centre of a relatively large grate than at the edges, probably in the ratio of 2 to 1.

The rate of combustion is of importance only in so far as it affects the complete combustion of the fuel and the temperature attained. That rate of combustion is the best which will enable the highest temperatures to be maintained in the furnace with the smallest possible consumption of fuel. The limiting rate is the highest rate at which fire can be maintained in good condition for the desired stoking interval.

If certain qualities of fuel are put on the fire and too high a rate of combustion is maintained without the fire being constantly attended to, clinker will form on the fire bars and become fused with the metal, thereby destroying the fire grate.

The conditions under which a heating boiler works are somewhat different from those of an ordinary power boiler, inasmuch as a heating boiler is expected to burn for a much longer period without attention than is a power boiler. It is quite a usual thing for a heating boiler to burn for six or eight hours without any attention whatever, and much longer periods even than this are not uncommon, whereas a power boiler in most cases must be attended to every half hour at least, and usually very much more frequently than this.

It is therefore clear that high rates of combustion can be maintained in the case of a power boiler, which would be impossible in a heating boiler, except under similar conditions.

The rates which are commonly adopted in power boilers are as follows :—

Kind of Boiler.	Pounds of fuel burnt per square ft. of grate.
Ordinary Cornish or Lancashire boiler, slow rate ...	4
Ordinary Cornish or Lancashire boiler	10
Ordinary boiler for factory purposes	12 — 16
Marine boilers	16 — 24
Greatest rate commonly possible with dry coal ...	20 — 23
Quickest rate for caking coals, where air is admtited above the furnace	24 — 27
Locomotive and torpedo boats boilers, or boilers with forced draught	40 — 120

With a heating boiler with solid bars a common rate is 4—6 lbs. per sq. foot per hour, while with waterway fire bars up to 20 pounds is not uncommon.

In cases where it is possible to adopt a high rate of combustion per square foot of grate, it is certainly economical to do so, because the higher the rate of combustion per square foot of grate the less air is practically necessary per pound of fuel in order to attain complete consumption, and therefore the higher the temperature which may be attained in the furnace.

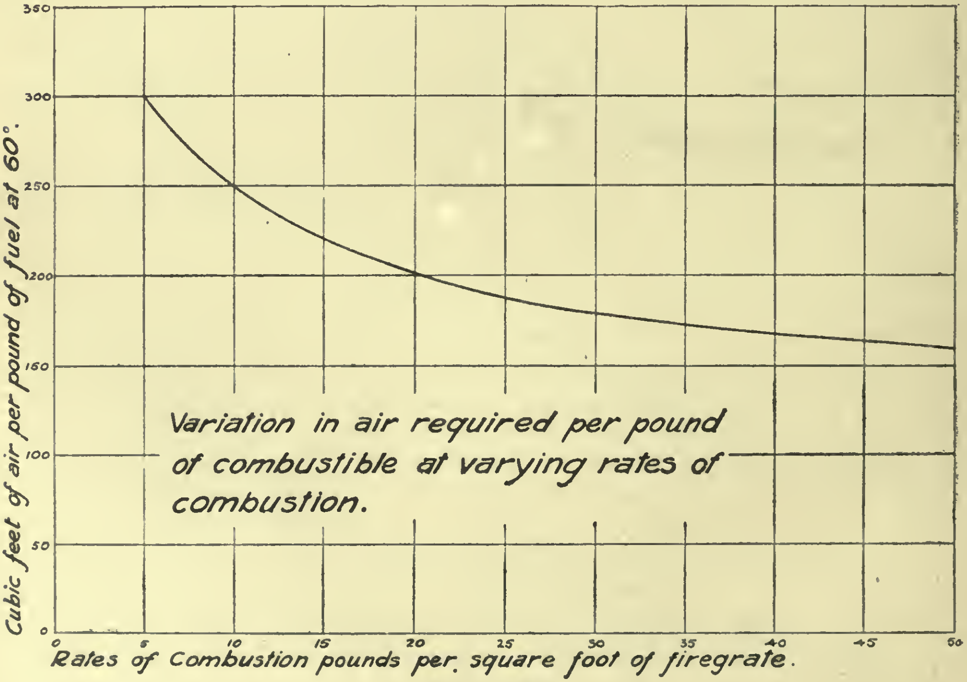


FIG. 104.

The diagram, Fig. 104, shows the number of cubic feet of air measured at 60° per pound of coal commonly necessary when different rates of combustion per square foot per hour are maintained on the fire grate with a suitable thickness of fire in each case. In one observation where the furnace temperatures were compared with natural and forced draught, with natural draught 24 lbs. of air per pound of coal consumed were drawn through the fire giving a temperature 2,926° F. The same furnace with forced draught burnt one pound of coal with 18 pounds of air, and the temperature produced was 3,686° F.

For ordinary calculations the amount of air commonly reckoned on for one pound of fuel is about 250 to 300 cubic feet at atmospheric pressure, or 24 pounds of air. The figures bearing on this point are given in the chapter on combustion of fuels.

The higher the rate of combustion the thicker it is necessary for the fire to be, and it is possibly partly the reason of the increased economy when higher rates of combustion are maintained, that thicker fires are kept. If too thick a fire is used with low rates of combustion, and if at the same time a high temperature is maintained on account of the heat generated not being drawn off rapidly enough, the air remains too long in contact with the incandescent fuel, and an excessive quantity of CO is produced, thereby wasting the fuel and reducing the efficiency of combustion.

On the other hand, if with a high rate of combustion the fuel bed is too thin, an excess of air rushes through without being properly carbonised, that is, without having its due proportion of oxygen converted into CO_2.

The temperature of the flue gases is thereby reduced and loss of efficiency is caused. The proper temperature in the furnace of a heating boiler is from 1,800° F. at low powers to 2,500° F. at high powers.

As explained in the previous chapter the meaning of the expression, "Temperature in the furnace," is open to considerable doubt. It is impossible to determine exactly the temperature of the gases apart from the effect of radiation and impossible to determine exactly how much heat passes to the plates by conduction and radiation respectively. It is known that when the interior plates of a furnace are shielded from direct radiation from the glowing fuel the rate of transmission is reduced by about 60%.

Experiments made with different thicknesses of fire throughout a considerable period of time with a total draught of ¾″ of water, gave the following result :—

Approximate thickness of fire.	Pounds of water evaporated from and at 212° per pound of coal.
9″	10.75
12″	11.25
14″	11.55

Economy of a Boiler.

The economy of a boiler depends on two features.

(1) The "temperature" attained in the furnace.

(2) The facility with which the heat is absorbed.

The effect of a good design of fire grate and chimney is to keep the temperature of the furnace high and thereby facilitate the conduction of the heat generated to the boiler plates. The effect of an ample quantity and a good arrangement of heating surface is to reduce the temperature at which the furnace gases are discharged into the chimney. The higher the initial temperature attained the smaller is the area of heating surface necessary to reduce the temperature of the gases to a given degree and the smaller is the loss of heat in the volume of gases rejected at the given temperature.

The diagram, Fig. 103, in previous chapter, shows the theoretical result which may be obtained by discharging the furnace gases into the chimney at different temperatures, on the assumption that all the heat not discharged into the chimney is utilised.

Consider these in order.

(1) It has already been explained that the temperature attained depends on the amount of air which it is necessary to supply to one pound of the fuel under given conditions in order to secure its complete combustion.

It has also been explained that the greater the rate of combustion on the grate, the lower in ordinary circumstances will be the proportion of air necessary to be supplied to each pound of fuel. The reason for this fact is not altogether clear.

It may be due to the fact that when high rates of combustion are maintained, it is possible and even necessary to have a thicker fire than where the rate is low. The air has a better chance of doing its duty, *i.e.*, in converting its oxygen into CO_2 than where the fire is thin and the rate of combustion low.

A thick fire combined with a low rate of combustion results in the air remaining too long in contact with the incandescent fuel, and the production of an abnormal amount of carbon-mon-oxide.

The economy of a boiler, therefore, depends partly on the possibility of maintaining a high rate of combustion and a thick fire. The higher the temperature of the furnace gases, the greater the ease with which they part with their heat to the boiler plates. The heat probably passes from the hot gases to the boiler plates partly by radiation and partly and perhaps chiefly by conduction.

(2) The heat given up to the boiler plates is obviously the total heat generated by the combustion of fuel less the heat carried away by the furnace gases.

In regard to the arrangement of heating surface, it is found by experiment that when the gases impinge on the heating surface normally, or at right angles, the surface is rendered much more efficient than when the hot gases flow parallel to the surface; doubtless the reason is the same as has been several times alluded to in this connection, that the effect of a surface placed in this relation to the stream of hot gases is to produce an efficient scouring action, whereby the layer of hot gas in contact with the cool surface is continually renewed. When the stream is flowing parallel to the surface there is a tendency for the same cool layer of gas to cling to the surface and therefore to reduce the actual difference of temperature between the gas and the surface.

Another feature which tends to increase the economy of a boiler is that the stream of cooling fluid should flow in a contrary direction to the stream of heating gases. When the gases are at their highest temperature they can easily communicate heat even to the hottest water, and the temperature of the water makes relatively little difference. When their temperature is relatively low, however, there is a marked variation in the temperature difference between gases and the hot water on the one hand and the gases and the cool water on the other, that is between the amount of heat communicated to a surface kept cool by relatively hot water and by relatively cool water. Therefore if the coolest gases are brought into contact with the coolest surface, it is possible to reduce the temperature of the gases to a point considerably below what is possible when the opposite conditions obtain, and sometimes in extreme cases actually below the temperature of the hottest of the water.

It is an extremely difficult experimental problem to determine what portion of the total heat is communicated to different parts of the heating surface. In the great majority of cases the mean heat-absorbing power of the surface is calculated over the whole boiler. The total heat abstracted from the fuel is measured, and this divided by the total heat receiving surface gives the average amount of heat absorbed by an average square foot of the surface. This is, of course, an exceedingly unsatisfactory method, and leads to large variations in the value of K.

It is found that where almost the whole of the heating surface of a boiler is direct surface and is directly exposed to the fire, or to the radiation from the hot fuel, the value as high as 7,000 B.T.U. per square foot per hour may be taken, or even higher, whereas when the surface includes, as usual, an amount of flue surface which only receives the hot gases after their temperature is considerably reduced, the over all rate is much lower.

The *average* B.T.U. reckoned on for sectional boilers of ordinary construction is from 2,500 to 3,000 B.T.U. per square foot per hour.

It is generally impossible to reduce these figures to a rate of transmission per square foot *per degree* per hour, as in the case of a radiator, because of the considerable difficulty of determining with any approach to accuracy the mean temperature of the gases. To determine this figure experimentally would involve separation of the heat absorbed from the furnace portion of the boiler from that absorbed by the flue portion, and a separate determination of the temperature and of the surface exposed to each temperature. The mean value of K for the entire boiler is assumed to be about 3.0 to 3.5.

The general rule is that when a boiler is working at its normal rate its proportions should be such that the gases are discharged into the chimney at a temperature not higher than 350° to 400° F., and an even lower temperature, down to, say, 280° F. is desirable in order to obtain maximum efficiency.

The power of a boiler, however, has to be considered when it is working not only at its normal rate, but also when it is being forced, that is to say, when the temperature is being raised as rapidly as possible.

In these circumstances the gases may escape at a higher temperature than when the boiler is working at its ordinary low rate, say from 600° to 650° F. as an extreme figure.

It is important that the boiler shall be so proportioned that it works at its maximum economy when the power is such as is required in ordinary circumstances.

A waste of power in exceptional or unusual conditions is of relatively small importance; what is of importance is that in normal circumstances the boiler shall be working at its maximum economy or efficiency.

On the Power of a Boiler.

It will be seen from the above discussion of the causes which produce variation in the amount of heat which can be delivered by a boiler, that the actual power of a boiler cannot be expressed in one figure without a good deal of explanation.

It has been shown that the number of B.T.U. which can be delivered by a given boiler varies with (1) the quality of fuel used; (2) the chimney draught; (3) the state of the boiler as to cleanliness, including in this term both the freedom of the flues, etc., from soot and deposit and of the interior of the boiler from scale, and the freedom of the fire grate from clinker; (4) the rate of circulation through the boiler; and (5) in a secondary or negative sense on the perfection of the coating applied to the outer surface of the boiler.

Since each one of these factors does actually produce a notable difference in the heat delivered, in accordance with the principles many times previously laid down, the power of the boiler cannot be correctly specified unless the value of all these conditions is specified also.

Further, for the engineer's purpose it is very desirable to know not only the power of the boiler in one set of conditions only, but also what alteration will be produced in the result when the conditions are varied.

Quality of Fuel.

Of the various factors on which the working power of a boiler depends, that relating to the quality of the fuel presents the greatest difficulty, for it is hardly possible to give what may be called a figure of merit of any particular fuel.

The number of B.T.U. transferred to the water from one pound of a fuel is not even proportional to the total number of B.T.U. actually in the fuel as given by a calorimetric test. The practical *power* of the fuel when burnt in a boiler furnace depends on many other qualities besides this, such as density, quickness of burning, closeness of texture, caking, smoke, soot and clinker producing qualities, character and quantity of ash produced, etc. All these qualities have an effect on the volume of air necessary to effect the combustion of one pound of the fuel, and hence on the temperature attained.

It is abundantly obvious that all these factors in the quality of the fuel cannot be taken account of by any scientifically determined coefficients, and that the only way of dealing with the fuel question is by strictly practical experiment. It is only possible to pronounce on the power of a fuel in a given boiler by actually trying it in careful comparison with other fuels in the same boiler.

Experiment alone can show that with a given boiler and a given draught a certain result can be secured with one pound of fuel of a certain quality which can be secured with, say, 1.23 pounds of fuel of another quality.

In order to give any approximate figure for the power of the boiler, so that boilers of different design can be compared, it is necessary, therefore, to specify one particular kind of fuel, which has a more or less constant quality, and to give the power of the boiler on the assumption that that fuel is used.

No doubt the best fuel for this purpose, that is, fuel of the most constant quality, would be a good quality of Welsh anthracite, of a given total calorific capacity, of the grade known as "Anthracite Nuts." But since anthracite is only used in a small proportion of installations, this would be somewhat unpractical.

It is advisable, therefore, to specify the power of the boiler on the assumption that a more generally used and therefore a better known fuel, although less constant in quality, is used. The quality of fuel in common use which comes nearest to a practical standard would be London coke, having, say, 12,000 B.T.U. per pound, the lumps having an average diameter of 2″.

The power of all boilers, therefore, should be specified on the assumption that fuel of this quality is to be used, although even with this specification there is room for considerable variation.

Rate of Combustion.

The grade of fuel having been fixed, the B.T.U. obtained from a boiler depends on the amount of fuel burnt on the grate, but not in proportion. The greater the amount of fuel burnt above a certain quantity, although the total emission may be increased, the smaller the proportion of the total heat generated is transferred to the water, other conditions being the same.

All heating boilers should be so designed that they need a minimum of attention, that is, so that they can be filled up with fuel and allowed to burn themselves out.

The higher the rate of combustion the shorter is the time during which one such charge will last, the greater is the rate of delivery of heat from the boiler, and the smaller is the proportion of the total heat generated by the combustion which can be transferred to the water.

In order, therefore, to specify completely the power of a boiler it would be necessary to give the power at several different rates of combustion, along with the time between the stokings necessary to maintain that rate.

The information required in order to specify the power of a boiler in practice cannot, perhaps, be more easily shown than by a diagram, such as Fig. 105. This diagram represents the results of an imaginary test of one boiler at six different rates of combustion. The time after the commencement of the experiment in hours is measured horizontally, each unit representing one hour. The total B.T.U. given off per hour are plotted as ordinates to the scale 1 unit = 20,000 B.T.U. per hour.

A preliminary consideration of the properties of this diagram is advisable.

Assume, for instance, that fuel is being burnt at such a rate as to generate 120,000 B.T.U. per hour, and that this rate of combustion proceeds for 15.2 hours. The combustion would be represented by a horizontal line, such as that marked *YP*.

Since each unit measured in a vertical direction represents 20,000 B.T.U. per hour, and each unit in a horizontal direction represents 1 hour, 1 square unit on the diagram represents

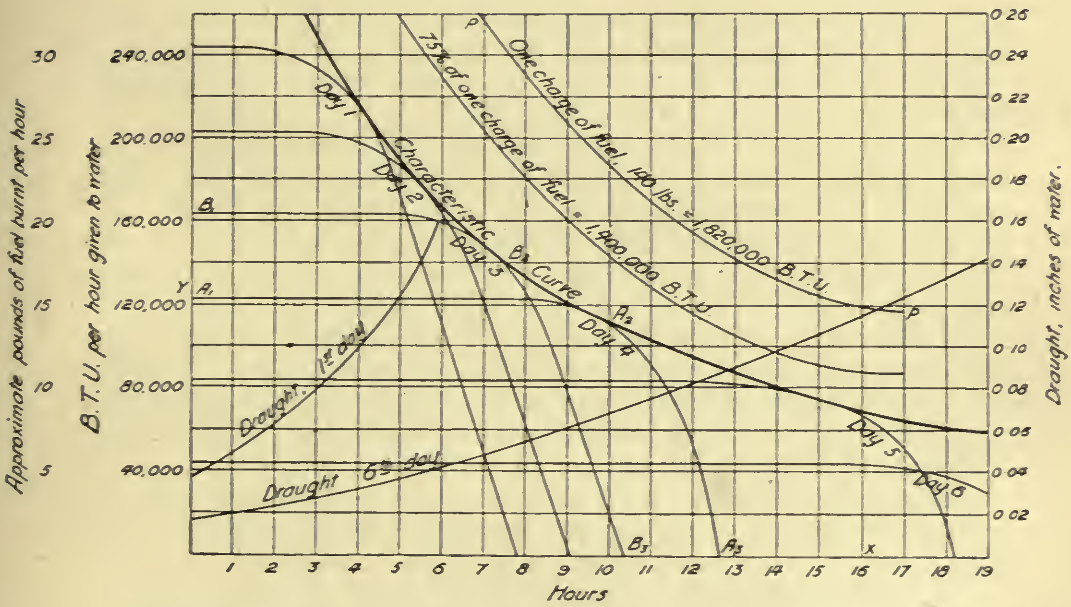
$$\frac{20{,}000 \text{ B.T.U.}}{1 \text{ hour}} \times 1 \text{ hour} = 20{,}000 \text{ B.T.U.}$$

A quantity of heat, therefore, would be represented on the diagram by an area, such for instance as *OXPY*, which would represent the total quantity of heat generated when 120,000 B.T.U. per hour was maintained for 15½ hours.

Assume that one full charge of fuel for the boiler under test weighs 140 pounds, each pound containing 13,000 B.T.U. The total number of B.T.U. in such a charge would be 13,000 × 140 = 1,820,000.

This charge might be burnt at a uniform rate of 120,000, 160,000, 200,000, or any other number of B.T.U. per hour. The corresponding time which would be occupied in completely burning it would be such that

$$\text{Time} \times \text{rate of combustion} = 1{,}820{,}000.$$



CHARACTERISTIC CURVE OF BOILER WORKED AT VARYING RATES.

A H Barker, delt.

FIG. 105.

Each rate of combustion could be represented by a different horizontal line.

All the points, such as *P* representing the end of the combustion, would lie on the rectangular hyperbola *PP*. A rectangular hyperbola has the property that the area under any point *P* on it such as *OXPY* is constant.

In this case it would represent 1,820,000 B.T.U., or in other words, it would be equivalent to

$$\frac{1,820,000}{20,000} = 91 \text{ square units,}$$

as shown on the diagram.

The curve *PP* therefore represents the whole of the heat derived from the combustion of 140 pounds of fuel.

When the fuel is burnt in the furnace of a boiler only a certain proportion, which may be taken as from 60% to 75% of the total heat generated, is communicated to the water, the proportion varying with the efficiency of the boiler and with the quality of the stoking and fuel, etc.

If careful tests are made on a boiler at different rates of combustion, the result of each such test might be represented by a curve in the diagram, such that the rate of heat emission to the water could be shown throughout the test. Such a curve is $A_1\ A_2\ A_3$. The drop in the curve near the end indicates that the fuel is nearly consumed.

The total amount of heat *communicated to the water* during the consumption of one complete charge is shown by the whole area $O\ A_1\ A_2\ A_3$ underneath the corresponding curve *AA*.

The efficiency of the boiler will then be the ratio of that area $O\ A_1\ A_2\ A_3$ to the area *OXPY* under a point on the corresponding hyperbola representing *the total heat* in one charge of fuel.

If the rate of delivery were increased to 160,000 B.T.U. per hour, the curve representing it would be $B_1\ B_2\ B_3$, and so on.

A set of diagrams of this kind would furnish complete information in regard to the heat using properties of the boiler when supplied with that quality of fuel.

If these figures are to be obtained by means of tests, each such test at one particular rate of combustion, should extend over one whole day at least, otherwise the figures obtained would not be reliable.

The draught necessary to maintain a given output would gradually increase from the time the test commenced towards the end, according to the condition of the fuel on the grate, in some such manner as is indicated by the draught curves on the diagram.

These curves show the draught which would be required at all times during the first and sixth days to produce the necessary output. The curves for the other days are not given, in order to avoid confusion on the diagram.

This variation of draught is produced by an automatic regulator, which is partly closed when the temperature is up to the desired value, and is opened wider and wider as the temperature falls.

The draught is assumed to be the difference in pressure under the fire grate and at the outlet to the chimney, and represents, as explained in the next chapter, the pressure required to draw the necessary quantity of air through the fire and flues of the boiler.

Efficiency on a Boiler.

The question of utilising the fuel *to the best advantage* is quite a separate one from that of obtaining the maximum power of a boiler under certain conditions.

The "efficiency" of a boiler is the average number of B.T.U. transferred to the water per hundred B.T.U. in the fuel burnt.

When very little fuel is burnt in the furnace the temperature maintained in the fire is so low that a large amount of air gets through the fire without being sufficiently carbonised, and the economy is therefore not good. As the rate of combustion is increased from a very low value the proportion utilised of the total heat in the fuel, or "efficiency," gradually increases up to a certain maximum. When the rate of combustion corresponding to the maximum efficiency is exceeded, the efficiency begins to diminish. When a high rate of combustion is attained the efficiency is materially reduced. These changes are shown on the diagram, Fig. 106.

This diagram plots as abscissæ the number of pounds of fuel burnt per hour, and as ordinates the B.T.U. utilised per pound of fuel.

The curves of efficiency representing the number of B.T.U. utilised per pound of fuel at different rates of combustion would be as shown at $Q_1\ Q_1\ Q_1$. The maximum efficiency in this case would be obtained when about 14 pounds of fuel were being burnt per hour, and at that rate the boiler would be delivering about 120,000 B.T.U. per hour.

A boiler of higher or lower efficiency would be represented by another similar curve, as shown dotted, higher or lower than the given curve.

It is clear that in a diagram such as this, if one unit of length, measured horizontally, represents, say, 2 lb. of fuel burnt per hour, and one unit of length, measured vertically, represents 1,000 B.T.U. utilised per pound of fuel, then one square unit on the diagram would represent

$$\frac{2\ \text{lb. fuel}}{\text{hour}} \times \frac{1{,}000\ \text{B.T.U.}}{1\ \text{lb. fuel}} = \frac{2{,}000\ \text{B.T.U.}}{\text{hour}}$$

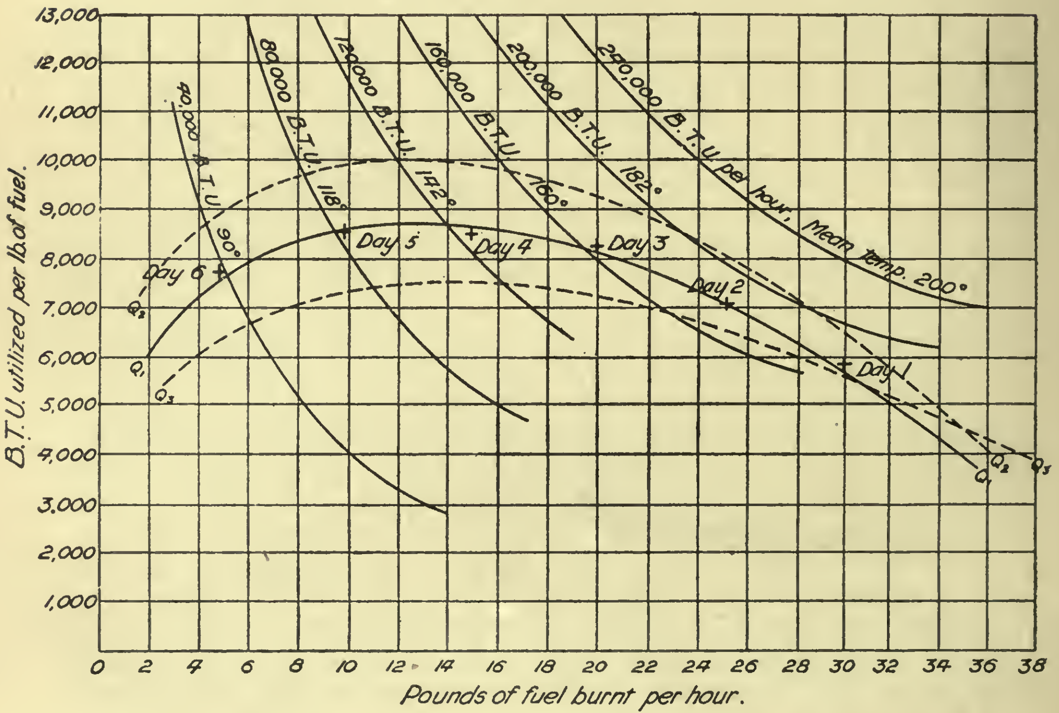
Thus, the area under any point on the diagram representing x pounds of fuel per hour, and y B.T.U. per pound of fuel, would represent xy B.T.U. per hour, or

$$\frac{x\,y}{2{,}000} \text{ squares on the diagram.}$$

All the points on the diagram, therefore, which represent a constant delivery of heat of xy B.T.U. per hour would lie on the curve $xy =$ constant, which is the equation of a rectangular hyperbola, the product of whose abscissa and corresponding ordinate is constant for all points on the curve.

Then if a boiler transmits 120,000 B.T.U. per hour, and if it burns a quantity of fuel per hour represented by *OX*, it is clear that the same boiler utilises per pound of fuel *OY* B.T.U. where *Y* is any point on the curve and *X* its projection on the horizontal base line.

Such a curve as this represents all the possible cases in which the total transmission is 120,000 B.T.U.



EFFICIENCY CURVES FOR BOILERS OF VARYING EFFICIENCY BUT SIMILAR SIZE.

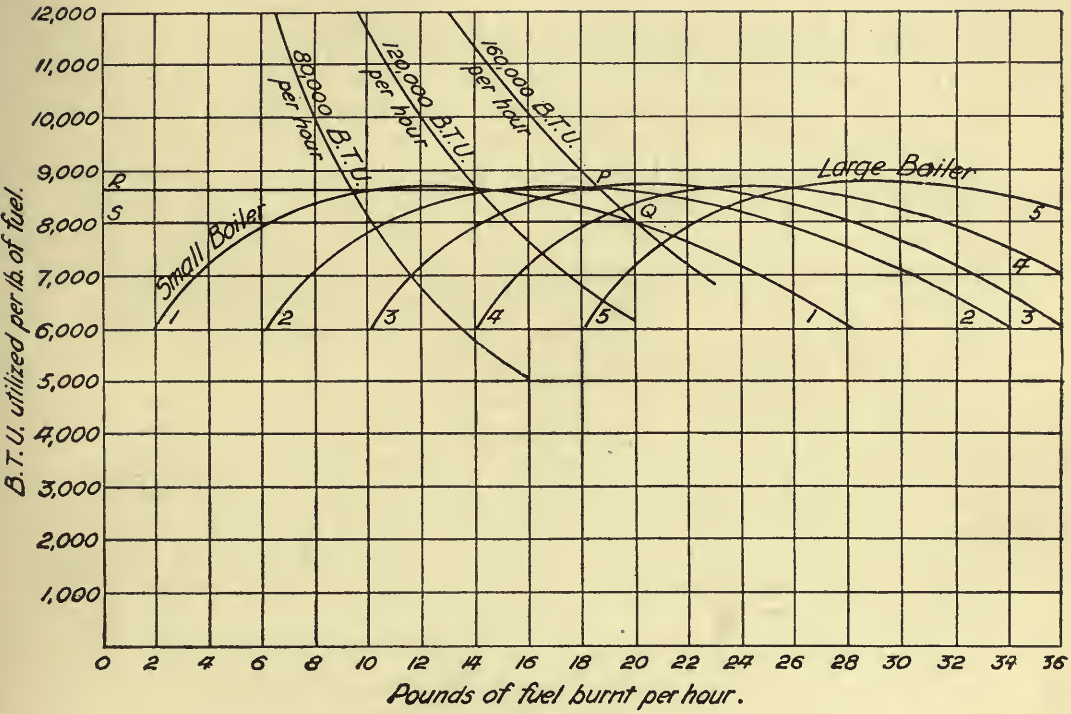
A.H. Barker, delt.

FIG. 106.

Let a series of such rectangular hyperbolas be drawn, each representing one particular rate of total transmission, cutting the curves of efficiency as shown. The length of the abscissa of the point of intersection of these hyperbolæ with the respective curves of efficiency indicates the quantity of fuel which it would be necessary to burn in each boiler in order to produce the result represented by the hyperbola.

Consider the point of maximum efficiency of the boiler, represented by curve Q_1 Q_1. This would require about 14 pounds of fuel per hour to maintain 120,000 B.T.U. per hour, whereas the boiler, represented by the higher curve Q_2 Q_2 Q_2, would only need 12 pounds per hour for the same duty.

On the other hand, the boiler of lowest efficiency, represented by the curve Q_3 Q_3 Q_3, would need about 16 pounds per hour.



EFFICIENCY CURVES FOR UNIFORM SERIES OF BOILERS OF VARYING SIZE BUT SIMILAR EFFICIENCY.

A.H. Barker, delt

FIG. 107.

A series of curves, therefore, such as Fig. 106, representing the efficiency of different boilers at different rates of combustion, would form a very complete indication of their respective economies at any given output.

A series of boilers of the same design and presumably of the same efficiency, but differing only in power, would be represented by a series of curves, such as Fig. 107, all of the same maximum height. A series of hyperbolas cutting these would show what advantage would be secured by supplying a larger or smaller boiler for the same duty.

It may be noted that each of the crosses on Fig. 106 is roughly calculated from the results of the experiments on the boiler indicated on Fig. 105, with the difference that the supply of fuel in Fig. 106 is supposed to be regular and continuous, and not as shown on Fig. 105, where each full charge of fuel is allowed to burn itself out.

Practical Aspects on the Rating of Boilers.

The question of the commercial rating of boilers used for heating systems is one of the most difficult of all those with which a heating engineer has to deal, on account of the wide variations between the power actually generated according to the conditions under which the boiler is installed, as already explained.

To the heating engineer it is a matter of the most vital importance that the power of the boiler shall be well proportioned to the amount of work it has to do, that is to say, that it shall neither be below power on the one hand, nor on the other that it shall be so powerful as to cause over-heating, and perhaps boiling of the water.

Up to within a few years ago it was the custom to specify the power of a boiler as "equal to" so many feet of 4″ piping. More recently still, the number of square feet of radiation which the boiler was supposed to supply was substituted for the number of feet of 4″ piping. Neither of these methods, however, can possibly give any accurate idea of the actual power of the boiler, unless at the same time the amount of heat which is assumed to be emitted by one square foot of the radiation is also specified.

It has been pointed out in Chapters XIII. and XVI. that the actual amount of heat emitted by any form of radiating surface depends wholly on the conditions. Even when the mean temperature of the water circulating in the pipe is specified, this alone does not enable the emission to be accurately determined. Thus, if fixed overhead near to cold glass in a single row, a pipe may emit twice as much heat as if it is fixed in a battery consisting of several rows in an underground channel. Such a method of specifying the power of a boiler is therefore obviously inaccurate.

If the number of square feet of radiation is to be specified the figure must also be given showing the number of B.T.U. assumed to be obtained from each square foot of the surface. The only object of giving these two figures is to multiply them together so as to obtain the total B.T.U. obtained from the boiler.

It is therefore entirely unnecessary to give two different figures when what is actually required is their product. The only correct and by far the simplest way of specifying the power of a boiler is the number of B.T.U. per hour which it will supply. But even so the actual power in B.T.U. obtainable from any boiler varies very much according to circumstances.

It is often possible, by altering the conditions under which a boiler is working, even to double the amount of power which can be obtained from it. There are, therefore, two things to be considered in specifying the power of the boiler, first, exactly what is meant by the power of the boiler, and, secondly, what are the conditions under which the boiler will deliver the amount of power specified.

It is obvious that so far as the heating engineer is concerned it is of no use to specify the power of a boiler under conditions which cannot be obtained in ordinary practical use, it is not even sufficient to specify the power of a boiler which a skilled stoker can obtain out of the boiler. The people who attend to the boilers in practical work are not skilled stokers. If satisfactory results can only be obtained from a boiler by the use of skilled attention, it is quite clear that the great majority of boilers will be unsatisfactory.

What is required in actual practice is the power which can be obtained from the boiler when properly erected, when provided with an adequate draught, and worked under such conditions as an ordinary intelligent caretaker can observe without an inordinate expenditure of time, trouble, and attention. There is not much difficulty in defining the conditions necessary for obtaining the *best* result. The difficulty consists in determining which of those conditions can be, and to what extent they ought to be observed by an ordinary caretaker. The whole result obtained depends very largely on the conditions which are actually observed.

There are many practical questions of essential importance which must be considered in connection with this matter.

Cleanliness.

Boilers differ very materially in the ease with which the flues in the fire grate can be kept clean from soot, dust, and clinker, so that if there are two boilers of equal power when perfectly clean, other things being equal, the boiler which can be most easily and thoroughly cleaned will, in practice, give better results than the other, or again, of two boilers, one might be of greater power with one kind of fuel, while with another kind of fuel the second one might be just as much in excess.

Chimney.

It has been indicated above that the chimney power is one of the most important considerations in relation to the power of a boiler. The question is treated in detail in another chapter.

The function of the chimney is to aspirate a sufficient, but not excessive quantity of air through the incandescent fuel, in order to obtain the combustion of fuel of the required quantity per hour. If this is done in such a way as to maintain the requisite temperature in the combustion chamber, then the furnace, as distinct from the boiler, is doing all that can be expected of it, and the chimney is not at fault. If in these circumstances there is a defect of power in the boiler, it is an indication of want of efficiency in the boiler, or that the circulation through the boiler is not good.

The circulation through the boiler is not an easy matter to investigate. If the quantity of water flowing through the boiler is known and the difference of temperature between flow and return pipe is also known, then the product of the two gives accurately the number of B.T.U. which the boiler is supplying. This requisite number of B.T.U. can be obtained either by increasing the temperature difference and reducing the flow, or vice versa. The only measurements which are easy to take are the difference of temperature between flow and return.

The quantity of water flowing is most difficult to estimate, and it is only by investigating the temperature in the pipes at various points that one can make up the heat account and ascertain where the heat generated by the boiler is escaping, and even this problem is frequently very difficult and very elusive.

In investigating a case of alleged defective boiler power a heating engineer should first ascertain whether the furnace is burning the requisite quantity of fuel, and whether the fuel is of proper quality, and, if possible, what is the temperature of the furnace. If satisfactory indications are given in regard to each of these matters there are only two explanations of any defect in power, one being the boiler is not efficient enough to transfer a due proportion of the heat generated to the water in the boiler, the second that a miscalculation has been made of the amount of heat emitted from the surfaces of pipes and radiators.

An indication in regard to the former may be obtained from the temperature of the outlet flue gases. In regard to the latter it should be recognised that a ventilating radiator over which a considerable quantity of air is passing loses a great deal more heat than does a radiator not so placed, and this is frequently a cause for miscalculation. Another fruitful cause is that the large amount of heat lost from the surfaces of exposed pipes is not duly allowed for.

There are two ways of increasing the power of a boiler, either (1) by burning more fuel in the furnace, or (2) by increasing the size or number of sections of the boiler. Of these the former is generally the cheaper to the engineer, and the latter is cheaper generally to the client.

The latter device will only increase the power of the boiler very slightly unless it enables more fuel to be burnt. If the defect in power is merely due to the chimney being too small to burn the requisite quantity of fuel, it is not of the slightest use to add sections to the boiler. This will only make matters worse.

The amount of heat obtained from a boiler may be increased by increasing the quality of the fuel supplied to it, and almost always by improving the attention bestowed on it, and by keeping the fire cleaner.

If the last causes a great increase in the amount of attention, it will generally be found impractical, for clients and owners will not generally consent to boilers being frequently attended to, and they are generally dissatisfied if the boiler furnace, after it is once attended to, will not last for at least three or four hours.

The general efficiency of a boiler of good design varies in practice between 60% and 75%. In special circumstances some boilers have an efficiency as high as 85%. This may probably be taken as the extreme limit in practical work. The meaning of the above statement is, of course, that for every 100 thermal units developed by the combustion of fuel in the furnace of an ordinary well designed boiler, from 60 to 85 thermal units will be transmitted to the circulating water or steam in the interior of the boiler, if the flues are kept in a moderately clean condition. Naturally the cleaner the flues are kept the better the efficiency.

If, therefore, it is ascertained by experiment that, say, 30 pounds of moderately good fuel per hour can be burnt in the furnace of a well designed boiler of any standard make, it is certain that, say,

$$30 \times 13{,}000 = 390{,}000 \text{ B.T.U.}$$

are being developed in the combustion chamber of that boiler per hour. On the assumption that the efficiency of the boiler is to be 60% (and each design of boiler has its own particular efficiency) and that the flues of the boiler are moderately free from dirt, soot, tarry deposit, etc., it is fairly certain that the amount of heat communicated to the circulating water or steam will be 390,000 × 60% at least.

In case the requisite quantity of fuel cannot be burnt in the furnace it is an indication either that the chimney draught is at fault, or that the resistance in the boiler is too high. Steps should be taken to investigate the cause. Any experienced engineer can tell from the appearance of a fire whether the draught is doing its duty or not, but it is impossible to tell whether the requisite quantity of fuel is being burnt in the furnace, necessary to produce the desired result otherwise than by weighing the fuel into the furnace and calculating exactly the consumption per hour.

The boiler troubles with which the heating engineer is confronted are many and various.

(1) What may be called merely mechanical or constructional troubles, consisting of leakage, cracking of sections and breaking of fire bars,

damage to doors, and fittings. These are matters of construction and erection and they concern the manufacturer or erector.

(2) Outside these obvious troubles there are troubles due to want of power or excessive power.

(3) Chimney troubles, due to poor or excessive draught or to down draught causing combustion troubles such as back-firing.

(4) Poor circulation in the circuit, resulting in the passage of insufficient water through and therefore a want of efficiency in the boiler.

Insufficient power in a boiler supplying an installation may arise from many causes. It should never be forgotten in determining the size of a boiler for a given installation that the boiler ratings given in makers' catalogues are made up under stress of competition.

The fact that makers are competing against one another in selling boilers puts a severe temptation on them to give catalogue ratings for boilers higher than they ought to be worked at if considerations of economy and fuel consumption are taken into account.

It may be taken as an axiom that the ratings in many, or perhaps most, catalogues of boilers are the very highest that the manufacturers can reasonably claim, and in some cases as high as they dare claim. It is, no doubt, also true that in most cases the catalogue ratings *can* be obtained if the conditions are sufficiently favourable. It would be too dangerous, from a commercial point of view, for manufacturers to give in their catalogues ratings which could not be obtained under *some* conditions.

The well-known makes of boiler have been supplied in such large numbers to the trade that the manufacturers have, presumably, experience of their power in actual working to show them that given certain conditions these ratings can be worked up to. If the boilers are reasonably rated these conditions should be easily obtained or not unusually favourable.

The more scientific firms of boiler makers rate their boilers as a result of elaborate tests at their own works. In other cases the ratings are fixed partly from calculation of the heating surface, which is a very uncertain basis, and partly, it is to be assumed, as the result of experience in actual working.

This also is a very elusive method of fixing the power of a boiler, for even an expert would find the greatest possible difficulty in determining definitely the power of a boiler from a test made on an actual plant, when it is not provided with the somewhat elaborate apparatus necessary for making exact observations.

It is the conviction of the present writer that no tests of a boiler can be relied on made otherwise than with a plant specially designed for the purpose. The uncertainty has been accentuated by the fact that hitherto the heating engineer has been content, in this country at any rate, to accept as a figure for the power of a boiler, either a number of square feet of radiation which the boiler is said to be capable of serving or the number of feet run of 4″ pipe, and the vitally important question of the mean temperature of the water in that surface has been overlooked.

It will be understood from the discussions on heat emission from radiating surface that such methods of specifying boiler power are wholly inadequate. The heat emitted from a 4″ pipe, which may perhaps be accepted as a standard form of radiation, varies widely according to the temperature of the water within it, and according to the position in which it is fixed. Such variations may easily amount to 100% for even moderate differences of temperature.

It will be seen, therefore, that to specify the power of a boiler as so many feet of 4″ pipe, or so many feet of radiation, means nothing at all, unless both the temperature of the water within the pipe and the position in which the pipe is fixed are specified along, of course, with the temperature of the surrounding air, and the fact that engineers have been hitherto content to accept such a specification as this is largely responsible for the very confused state of the matter of boiler rating at the present time.

The only correct way of specifying the power of a boiler is to give the number of B.T.U. the boiler is capable of delivering into the pipes under given conditions, and the specification of these conditions is the cause of the whole difficulty.

It has been shown in the previous discussion that a boiler may be worked at a wide range of different rates. Which of these rates is to be accepted as the standard?

A suggestion has been made that the rate can be given in terms of the number of hours which a full charge of fuel will last in the boiler. But this method is unsatisfactory, because it causes confusion between two wholly separate features of the boiler, namely :—

(1) The B.T.U. which can be obtained from it, and
(2) The size of the fuel space or magazine which the boiler contains.

These two features of a boiler are wholly distinct. The B.T.U. which can be obtained may be taken as very roughly proportional to the square feet of heating surface, while the length of time which one charge of fuel will last depends roughly on the cubic contents of the fire pot, and between these two there is no sort of proportion possible. The relation between them depends entirely on the design of the boiler.

In order, therefore, to get the power of a boiler with any approach to accuracy, the information conveyed on some such diagram as Fig. 105 is necessary, and it has been explained that even a diagram of this kind only applies to one particular kind of fuel.

The most satisfactory method of specifying the power would be to draw what may be called a characteristic curve for the boiler in question, passing through the points of the curves, Nos. 1 to 6, in the manner shown. This curve would give not only the power of the boiler in all circumstances, but also its efficiency, by comparison with the curve showing the total heat in the fuel consumed.

For ordinary commercial purposes it is necessary that the nominal power of the boiler should be specified as one figure, and therefore one of these curves must be selected as representing the power of the boiler. The curve selected may be such that one charge of fuel will last a given time, say, six hours, but it must be clearly understood that this figure would only convey a fairly accurate idea of the power of the boiler for one particular type of boiler, and for a type which had been specially designed for a six hours' run.

Between boilers varying so widely from one another as, say, a sectional boiler and a cylindrical wrought iron boiler, such a figure would be quite misleading, for a cylindrical boiler is not commonly designed to last for so long as six hours. In fact, such a figure would represent the power of a boiler when the boiler had been specifically designed for that length of run.

On the other hand, with a magazine boiler, which is designed to run for a longer period than six hours, the figure would be also widely in error. However, since cast iron sectional boilers are now by far more commonly employed than any other type, and since most of them are designed for a six hours' run, such a figure might be taken as being fairly satisfactory for that type, provided always it was designed for a six hours' run.

It should also be incumbent on the maker of the boiler to specify the intensity of draught which is necessary throughout the run to maintain this rate of output.

In any case there is some uncertainty as to what is meant by a six hours' run. If some intermediate attention to the fire is permitted, not only would the necessary draught be much smaller, but also a better output might be obtained. Any fire when it is left for a long period like six hours becomes partially choked up with ashes.

Shaking grates have been introduced, whereby the fire may be stirred up at intervals without considerable trouble, so as to clear the grate of small ashes and dust. But these grates have not found much favour in this country, owing to their perishable nature and their tendency to get overheated and destroyed, and also the tendency of small bits of fuel and hard ashes to clog up the grate so that it will not work.

No proper investigation of a boiler plant is possible if only the square feet of heating surface carried by it are known. This will be understood by reflecting that if a square foot of heating surface is maintained at 170° it emits about twice as much heat as the same square foot at 130°. The maintenance, therefore, of a given number of square feet of radiating surface of 170° would need a boiler exactly twice as large as the maintenance of the same installation at 130°.

In both these cases the radiators would feel hot to the hand, and the difference between the temperatures, while it would be noticed of course by an expert, might easily escape the attention of a relatively inexperienced person. This instance illustrates the futility of rating boilers by the square feet of radiation.

Given the number of B.T.U. which the boiler should develop, it is next necessary to calculate approximately the number of pounds of fuel per hour which should be burnt to produce this result. This calculation is merely an application of the following formula:—

$$\text{lbs. fuel} = \frac{\text{B.T.U.}}{8{,}000}$$

If the boiler is clean and if it can burn the required quantity of fuel of the proper quality, this is evidence that the chimney is not at fault. It may, of course, be producing less draught than it should, owing to leakages and other causes, but at any rate the chimney is adequate for its duty.

It is generally possible for an expert to tell from the appearance of the fire whether the chimney is doing its duty or not. If the required quantity of fuel is being burnt, and if at the same time the requisite heat is not being developed, the cause may be one of several.

(1) The boiler is not sufficiently powerful to utilise properly the fuel which is being burnt. If this is the case it will be proved by the fact that the temperature of the gases escaping into the chimney will be too high. If these gases are above 400° F. when the boiler is working at normal power it may be taken for granted that the boiler heating surface is not as large as it ought to be.

But this inefficiency may be caused by relative sluggish circulation through the boiler. Sluggish circulation is proved by the existence of too great a difference of temperature between the boiler flow and boiler return. If this temperature is in any case greater than 50°, and if at the same time the load on the boiler is not too great, the circulation may be known to be sluggish.

A high difference of temperature between the flow pipe and the return pipe may be due either to poor circulation or to excessive load on the boiler, or both.

If the circulation is good, and if at the same time the boiler is too small for its work, the effect would be indicated in general by the impossibility of raising the temperature of the return pipe, although the flow pipe is at a high temperature. But it by no means proves that the circulation is good if the temperature of the return pipe is reasonably near the temperature of the flow pipe.

The temperature of a return pipe may be raised to a great extent by a short circuit through some radiators or coils near to the boiler, all the distant parts of the pipes being cold.

The criterion of a uniform circulation throughout a system is the existence of the same temperature in all the return pipes. The flow pipes will generally take care of themselves if the return pipes are at the correct temperature.

Perfect circulation in a system exists when the temperature of all the return pipes is substantially the same as the temperature of the return main. If, however, it is possible definitely to locate the existence of boiler troubles in the chimney, it is necessary to ascertain to what cause this inefficiency is due. This matter is dealt with in next chapter.

On Steam Boilers.—Priming.

The essential causes of priming in steam boilers are as follows :—

(1) Impure water, containing oil or alkaline matter in solution. Some kinds of boiler composition are apt to produce very bad priming.
(2) Too small steam space.
(3) Too small free steam disengaging water surface.
(4) Lack of adequate circulation in the boiler, producing differences of temperature, and pressures in different parts of the boiler.

There is a rough general rule which is sometimes adopted, whereby the velocity of the steam given off from the surface of water normal to that surface should not exceed 2.5 feet per second. In other words, that not more than 2.5 cubic feet of steam should be disengaged per second from one square foot of water surface. This, of course, is equivalent to 150 cubic feet of steam per minute, or 9,000 cubic feet per hour, which at 5 pounds per square inch is equivalent to 455 pounds per hour.

This is probably an extremely high value, and if the space is smaller than this it is sure to produce violent priming. A more usual rule is to limit the emission of steam to 40 lbs. per hour per square foot. But the extent to which a boiler will prime depends wholly on its design, and appears to be little understood.

The conditions that favour dry system delivery from a boiler are facilities for easy circulation within the boiler itself, so that all parts of the boiler shall be at the same temperature, and so that there shall be no globules of steam carried downwards into the base of the boiler by the descending current. That is to say, that the descending current shall be clear of the influence of the fire as far as possible.

CHAPTER XX.

ON CHIMNEYS AND THE CONTROL OF DRAUGHT.

GENERAL PRINCIPLES.

THE function of a chimney is to draw such a quantity of air through the fuel on the grate of the boiler as will maintain combustion at the required rate per hour.

A chimney draws air through a fire by reason of the fact that warm flue gases are specifically lighter than cold air. The air rises within the chimney for substantially the same reason as that for which any mass of substance lighter than water when immersed in water will float to the surface.

The theory of the aspirating power of a vertical flue has been explained on page 195 in connection with the discussion of ventilation.

If too large a quantity of air is drawn through the fuel, the result is, of course, that more fuel than necessary will be burnt. If too high a rate of combustion is maintained by an excessive draught the fuel is not only burnt in an uneconomical manner, but there is a danger that the boiler grate may be damaged by overheating, generally owing to the formation of clinker and especially if the fire is not constantly attended to.

If, on the other hand, for any reason a chimney has not sufficient power for its work, it will fail to draw sufficient air through the fuel, and the requisite quantity of heat will not be generated. It is therefore very important that a chimney should be of suitable power.

It is more important, however, to ensure that it should be at least large enough than that it should not be too large. The draught due to a chimney which is too large can always be reduced as far as necessary by suitable dampers. If the power is too small it cannot generally be increased without considerable trouble and expense.

The total intensity of draught or circulating pressure forcing air up a chimney is the difference between the pressure of two columns of gas, each of the height of the chimney, the one of air at the temperature of the external air, the other of smoke gases at the mean temperature of the interior of the chimney.

In this case the strict meaning of the mean temperature t_m would be such that

$$\frac{H}{1 + a\,t_m} = \int_0^H \frac{d\,h}{1 + a\,t}$$

For practical purposes, however, the value of t_m is generally taken as the arithmetic mean of the inlet and outlet chimney temperatures.

The amount of air that will find its way through a chimney having a given circulating head, depends on the total resistance to flow of air of the boiler grate, flues, and the chimney shaft, including all the passages from the point where air enters the base of the boiler up to the point where it is discharged into the open air at the top of the chimney.

It is clear, since all and only the air which goes into the furnace passes out of the top, that in any airtight chimney the total weight of air passing any section of the chimney, or the flue, or the grate, or any other part of the system in a second, must be absolutely constant at all sections.

For if not, assume there are two sections across which the mean rate of flow in pounds weight per second is different. The flue gases must either accumulate in or be continuously abstracted from the space between the two sections. This is obviously impossible. The *volume*, however, of this equal weight of air changes from point to point, according to its temperature.

In other words, the volumetric velocity, that is, the number of cubic feet per second, is not constant, but depends on the temperature at the particular point. The greater the temperature at any section, the greater the mean velocity for that section.

The ratio between the volumetric velocity at any two sections where the temperatures are t and t respectively, is obviously

$$\frac{1 + a\,t_1}{1 + a\,t_2}$$

Magnitude of Total Draught Generated by a Chimney of Height *h*.

Let D = weight of 1 cubic foot of air at 0° F. = 0.0863 lbs.
Let D_i = weight of 1 cubic foot of flue gases at 0° F.
t_o = outside temperature in degrees F.
t_i = the mean temperature inside the chimney.

The pressure in pounds weight per square foot, due to a column of outside air of height h feet at temperature t_o° F., is then the weight of h cubic feet of air.

$$\frac{h \times D}{1 + a\,t_o}$$

The weight of a similar column of flue gases at a temperature t_i is similarly

$$\frac{h \times D_i}{1 + a\,t_i}$$

Hence the circulating pressure, assuming $D = D_i$

$$= h\,D\left(\frac{1}{1 + a\,t_o} - \frac{1}{1 + a\,t_i}\right)$$

pounds per square foot.

The same pressure expressed in inches of water is

$$h\,D\left(\frac{1}{1 + a\,t_o} - \frac{1}{1 + a\,t_i}\right) \times 0.192$$

or in feet of flue gas column

$$\frac{h\,D}{D_i}\left(\frac{1}{1 + a\,t_o} - \frac{1}{1 + a\,t_i}\right)$$

The density of the flue gases D_i actually varies from 1 to 1.08 of the density of air. For present approximate purposes it is sufficient to reckon the density as the same as the density of air at the same temperature, or

$$\text{Draught in feet of flue gas} = h\left(\frac{1}{1 + a\,t_o} - \frac{1}{1 + a\,t_i}\right)$$

The power of a chimney should always be calculated for the most unfavourable conditions. These conditions are a relatively high outside temperature, and a relatively low chimney temperature. When the outer temperature is at or above 60° as a maximum, a heating apparatus in ordinary circumstances is not working at all. Therefore, if 60° is assumed as the outer temperature, this is the most unfavourable possible for the chimney.

The lowest chimney temperature, when the apparatus is working at its lowest power will be in the neighbourhood of 300° F.

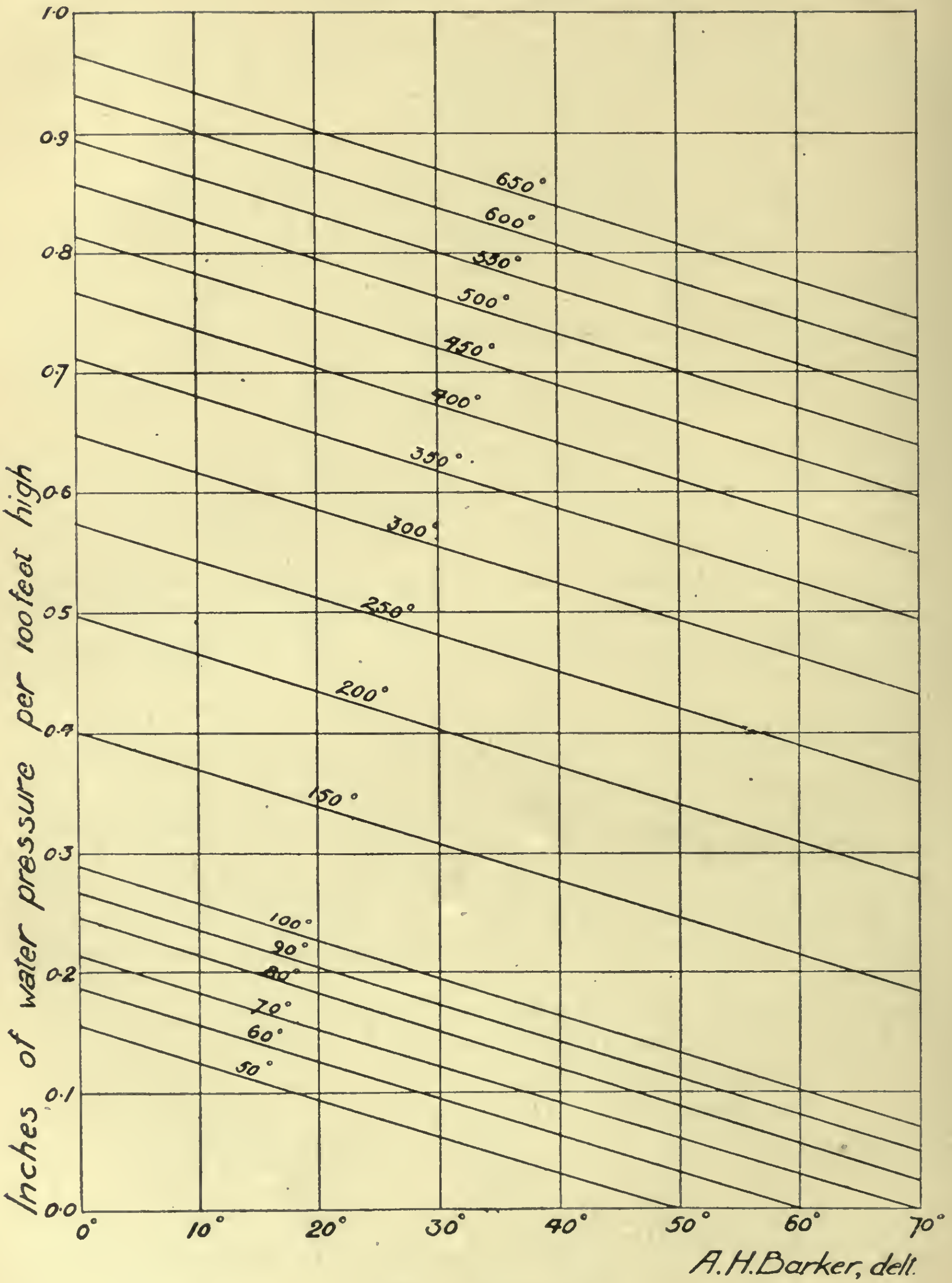
The above expression therefore becomes

$$h\left(\frac{0.0863}{1 + \frac{60}{461}} - \frac{0.0863}{1 + \frac{300}{461}}\right) \times 0.192 \text{ inches of water}$$

$$= h\left(\frac{461 \times 0.0863}{521} - \frac{461 \times 0.0863}{761}\right) \times 0.192 = h\,(.0767 - .0523) \times 0.192$$

$$= h \times .0244 \times 0.192 = h \times .00467 \text{ inches of water.}$$

The diagram, Fig. 108, gives the total circulating pressure per 100 feet



Air temperatures.

FIG. 108.

Circulating pressure produced by vertical shaft 100 ft. high.

of height, due to a chimney with different temperatures of the flue gas and external air. Roughly, a chimney 100 feet high may be taken to produce a circulating pressure of ½″ of water at lower power.

Alteration of Temperature within a Chimney.

If a chimney is airtight, the loss of temperature between the point where the flue gases enter the chimney and the top depends on the weight of flue gases per hour passing up the chimney, their temperature, and on the amount of heat lost from the substance of the material of which the chimney is made.

If the gases enter the chimney at a temperature t_1 and leave it at a temperature t_2, the drop in temperature is $t_1 - t_2$. The mean temperature within the chimney may be taken as

$$\frac{t_1 + t_2}{2}$$

Let K be the value of the co-efficient of transmission.

Let C be the mean circumference of the chimney. The surface losing heat is therefore hC square feet.

The amount of heat lost may therefore be taken as

$$h\,c\,K \left\{ \left(\frac{t_1 + t_2}{2} \right) - t_0 \right\} \text{ in B.T.U. per hour.}$$

If Q is the number of pounds of flue gases passing up the chimney per hour the loss of temperature will therefore be

$$\frac{h\,c\,K}{Q \times S} \left(\frac{t_1 + t_2}{2} - t_0 \right)$$

where S is the specific heat of the flue gases.

This expression is, of course, only an approximation. An accurate solution of the problem would be unnecessarily complicated.

In practice the loss of heat from a chimney can be disregarded when it is made of brickwork. It is not considered safe, however, where the chimney is made of metal. Assume that the chimney is h feet high and d feet in diameter. That loss can then be calculated from the formula

$$t_1 - t_2 = \frac{h\,\pi\,d\,K}{Q\,s} \left(\frac{t_1 + t_2}{2} - t_0) \right)$$

The value of K may be taken from the diagram, Fig. 108*a*, which represents the amount of heat transmitted through a thin metal plate in different conditions of temperature and velocity.

This calculation is of importance as determining the mean temperature of the chimney, and it is the mean temperature which must be taken in calculating the draught-producing power of a chimney according to the above formula.

Utilisation of the Total Draught Generated in a Chimney.

The total circulating power of a chimney generated as explained above, is utilised in drawing air through the fire and system of flues and shaft. The amount of air drawn through by this total head depends on the joint resistance to flow of air of the entire system of flues, including—

(1) The head necessary to generate and maintain the velocity with which air flows through the inlet door underneath the fire grate.

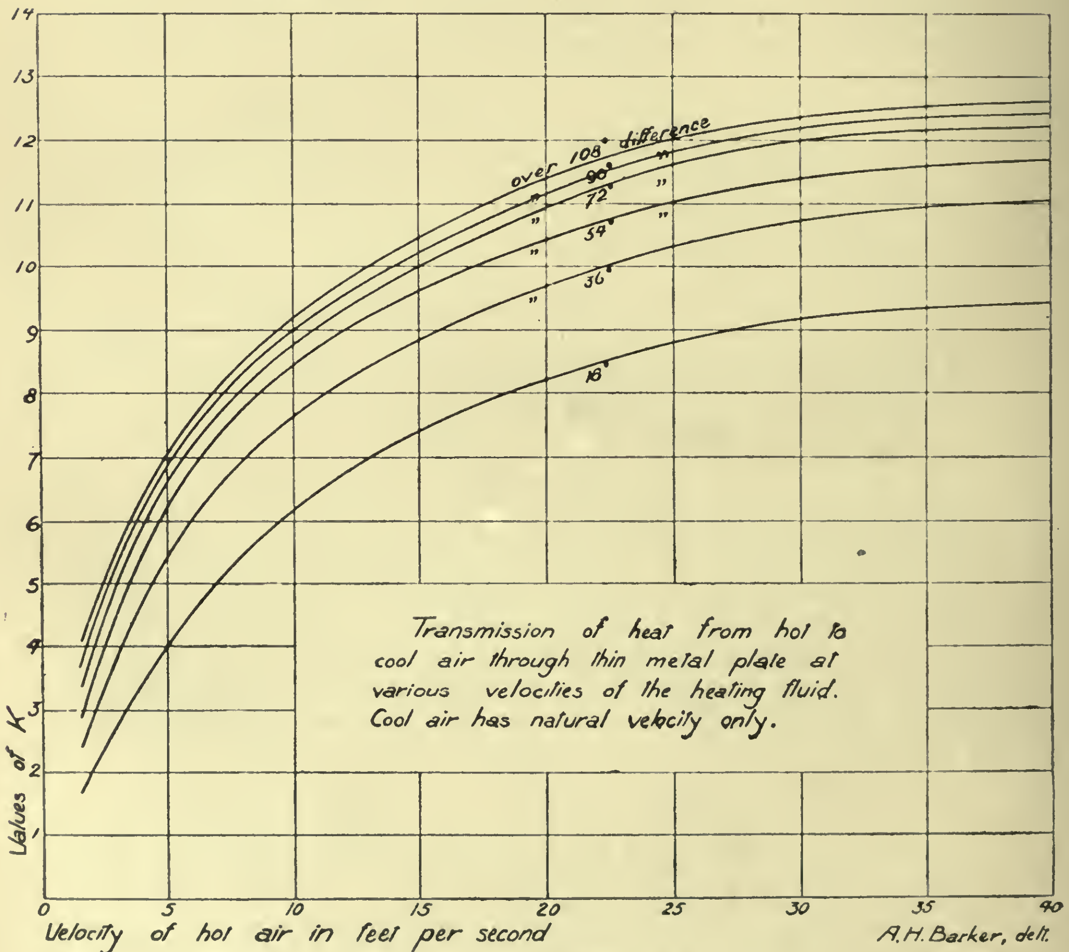


FIG. 108a.

(2) The resistance due to the passage of air through the grate and fire.
(3) The resistance due to passage of the flue gases through the flues of the boiler.
(4) The resistance due to the velocity generated in passing from the flues into the chimney.
(5) Friction in the chimney itself.
(6) The loss of head due to change of velocity with which the gases are discharged from the chimney.

In accordance with the principle which has been fully explained in another part of the present work, the forces of nature will so adjust the velocities in the various parts of the system that two conditions are rigidly observed:—

(1) The total weight of air passing across any section of the flues in a second is the same at all sections.
(2) That the sum of the whole of the resistances enumerated above shall be absolutely identical with the total head generated by the chimney.

The velocities attained in all parts of the system represent Nature's automatic solution of an infinite series of simultaneous equations, each of a high degree of complexity.

In order that it may be possible to calculate approximately the quantity of air which will be drawn through a boiler and system of flues in any given circumstances, it is absolutely necessary to know the resistance of the various parts of the system, and since some of these vary widely from time to time it is impossible that they can be generally known with accuracy.

Any tables, therefore, which purport to give the height and diameter of chimney necessary for any given duty are essentially incorrect, or at least they can only be regarded as rough approximations. If they are suitable for general employment in practice they must of necessity give sizes much larger than generally necessary.

The part of the resistance which varies most is, of course, that of the fire grate, and the mass of incandescent fuel on it. The resistance of this part of the system can vary as much as 500% or more, according to the condition of the fire, the size of coal which is being used, the manner in which it is heaped on to the furnace, and other matters.

The resistance of the flues of the boiler and of the chimney may also vary very considerably, according to the amount of soot which has collected in them, to the extent of 100% or 200%.

There is a still further complication introduced by the method which is commonly employed for regulating the draught. That method, which will be treated in a later section of the present chapter, is almost invariably the mechanical reduction of the area through which the air finds its way to the grate or into the chimney, or both.

It can be seen, therefore, that the actual resistance due to the passage of air varies greatly, and any pretence at exact calculations on the point is somewhat absurd in view of the practical conditions.

In order to acquire a correct conception of the flow of air up the flue when the resistance varies, the following explanation should be studied.

It is assumed that throughout the changes in the draught up the flue the interior and exterior temperatures remain constant—a condition which will in practice obtain approximately. The mass of heated brickwork prevents rapid changes in flue temperature.

What has been described as the total circulating pressure of the chimney will therefore remain approximately constant for a considerable interval of time. The alteration in the flow of gases up the chimney will be effected by changing by mechanical means the resistance of some part of the system of flues. In the great majority of cases this part is the inlet of air under the base of the boiler.

That part of the loss of head caused by the flues and stack remains constant only so long as the quantity of air passing up the flue remains constant. This portion of the resistance generated or loss of head varies approximately as the square of the velocity.

Let the vertical axis of Fig. 109 represent diagrammatically the various parts of the system of flues developed along a straight line. The height of the curves on the left of the figure above the base line represents the manner in which the total circulating head is employed.

So long as the temperature in the flue remains constant, the line which represents the total available circulating pressure also remains constant. This total available pressure is divided between the various parts of the flue, as shown by the curves.

Let the lower curve represent the loss of head throughout the flue at the various points in the system when the inlet at the base of the boiler is wide open.

The *relative* values of the resistances in the constant portion of the flues remain constant. Thus, if the velocity in any portion of the flue is doubled the velocity in every portion of the flue must be doubled also, and therefore the head lost in each portion of the flue is increased in the proportion 1 : 4.

Thus, the general shape of the curve remains the same whatever the absolute value of the flow of air may be, the heights of the curve at given points being altered in proportion. When the inlet is partially closed, the only portion of the curve which is altered in shape is that near the beginning.

If the inlet area is reduced, the pressure drawing air through it, and

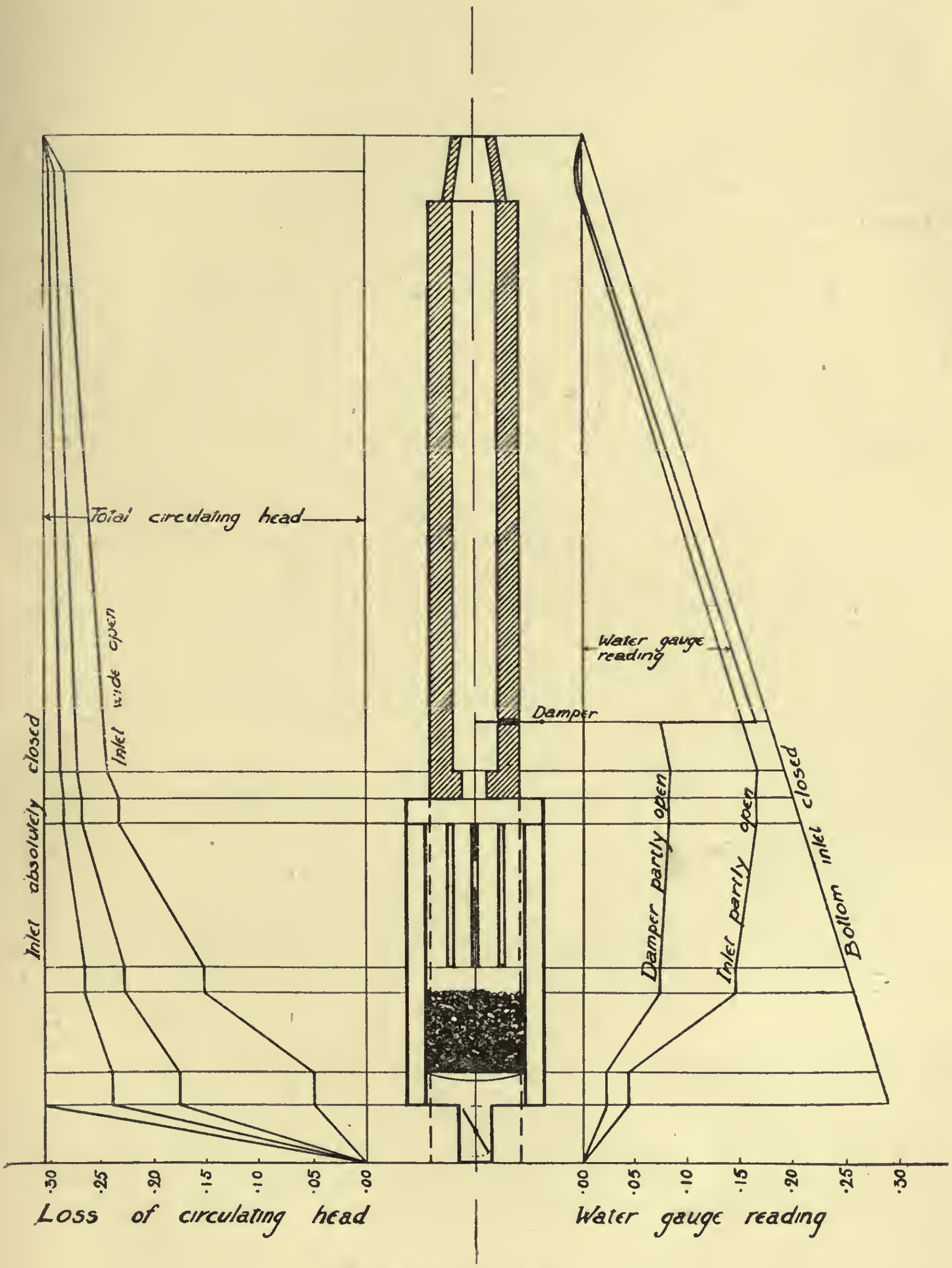


FIG. 109.

Circulating pressures and loss of head in a chimney.

therefore the velocity of the air is thereby increased. A greater proportion of the total circulating pressure is concentrated on forcing air through the reduced inlet, so that the amount of air is not reduced in proportion to the reduction of area. The difference in pressure is proportional to the square of the velocity through the opening.

If the area is reduced so far that the velocity of air through the opening is doubled, this would involve an increase in the difference of pressure between the outer air and the under side of the boiler grate in the ratio of 1 : 4. This additional loss of head causes a reduction in that portion of the total head which is available for overcoming the resistance through the fire and flues. So long, therefore, as the fire remains in a constant condition the actual delivery of air would be reduced in the proportion of the square roots of the differences of pressure on the two sides of the fire, so that the quantity of air would be only slightly reduced, although the area of the inlet were reduced by more than one half.

As will be shortly explained, it is substantially for this reason that secondary regulation by means of admitting air into the chimney is necessary, namely, that the regulation by partial closing of the inlet only, does not sufficiently reduce the volume of air passing.

The manner in which the pressures in the chimney go down when the inlet door is closed, are illustrated by the successive curves on the figure.

It will be observed that the greater the proportion which the resistance of the fire bears to the whole resistances the greater is the effect which the closing of the door has on the total flow of air.

If the whole of the circulating pressure, say ½″ of water, were concentrated on producing inlet velocity, as when the inlet at the base of the boiler is very restricted, the air velocity generated would be about 47.5 feet per second.

On the right of the figure the curves denote the actual suction at the various points of the system, as shown by a water gauge, and illustrate the effect on the water gauge reading of a partial closure of a fixed damper in the chimney.

Draught Necessary to Produce Combustion.

The determination of the magnitude of the circulating pressure or draught necessary to maintain a boiler fire is a very difficult question, seeing that the value varies very much according to the actual condition of the fire, and also varies very largely according to the quality of the fuel used.

As explained above, the total head due to the pull of the chimney is only partly employed in the furnace itself.

The relative proportions which the different portions of the resistance bear to one another varies very largely, according to the condition of the fire and according to the quality of the fuel used. The finer the fuel and the thicker the bed, the greater is the resistance to the passage of air through it, and the more difficult the combustion.

The following instance, taken from experiment, gives an approximate idea of the ordinary proportions in which the total available head is distributed into the six sections, where the fire of small coke is about 12″ thick, and where the total circulating pressure is about 0.3″ water gauge :—

Entrance under fire grate	.05″
Resistance of fire	.105
Boiler flues	.08
Flues into chimney	.01
Friction in chimney	.04
Discharge velocity	.02
	.305

It is not infrequently the case in a heating installation that the fire is considerably thicker than this, and in this case the resistance of the fire is proportionately greater. With larger fuel, on the other hand, the resistance would be less.

The following table shows the relative rates of combustion for different sizes of coal with the same draught :—

Size of Fuel.	Relative Rates of Combustion with Constant Draught.
Between 7/8ths and 9/16ths	100
,, 9/16ths ,, 3/8ths	85
,, 3/8ths ,, 3/16ths	70

The following table (derived from experiment) illustrates how the pressure must increase as the size of the coal decreases, in order to maintain approximately the same rate of combustion :—

Maximum size of coal.	Air pressure in inches of water.	Combustion per sq. ft. of grate per hour.
7/8	0.375	13.63
9/16	0.5	13.58
3/8	0.625	11.4
3/16	1.04	11.34

The following table gives an approximate idea of the relative draught required for different kinds of fuel, where ordinary coke, 2″ in diameter, is taken as the standard :—

Wood	0.6
Sawdust	0.7
Peat	0.8
Heavy peat	1
Coke	1
Ordinary steam coal	0.8—1.4
Slack	1.2—1.8
Very small slack	1.4—2.2
Coal dust	1.6—2.2
Semi-anthracite	1.8—2.4
Mixture of breeze and slack	2—3
Anthracite slack	3—3.5
Anthracite nuts	2.4—2.8

Size of Chimney.

It will be evident from the foregoing discussion that an accurate determination of the size of a chimney for any given duty is so difficult as to be almost impossible.

The principle which governs the size of the chimney is primarily that it must be able to exert a pull at the back of the boiler sufficient to draw the requisite quantity of air through the fuel.

This volume of air is determined from the quantity of fuel which it is desired to burn, and this again is given by the heat which the boilers are required to give off, divided by the heat transmitted per pound of fuel burnt.

For this calculation it is convenient to reckon that at the maximum power the boiler will utilise 8,000 B.T.U. of the total heat in one pound of fuel, and that the combustion of every pound of fuel produces 300 cubic feet of flue gases. If, for instance, an installation requiring 400,000 B.T.U. per hour is in question, it would suffice to reckon on a fuel consumption of

$$\frac{400{,}000}{8{,}000} = 50 \text{ pounds of fuel per hour.}$$

And the volume of these flue gases would be

$$50 \times 300 = 15{,}000 \text{ cubic feet per hour.}$$

Now the chimney must be such that when the loss of head due to passing this quantity of air per hour is allowed for, there must remain a surplus pressure sufficient to draw the same quantity of air through the boiler grate with the fuel heaped up, and through the boiler flues. If it will not do so, it is not sufficiently powerful. It is clear, therefore, that the power of the chimney depends on the resistance of the boiler when it is heaped up with fuel, and boilers vary considerably in their resistance.

Without any special information, therefore, as to the resistance of the boiler, it is desirable to allow for a high resistance boiler, and there are few boilers which will not pass the maximum quantity of air desirable in a heating boiler with a pressure of 0.1 inches of water. This implies that no chimney for a heating installation can be relied upon if it is less than about 16 or 20 feet in height, although lower chimneys than this have been known to give quite good results when the conditions are suitable.

Temperatures in the Chimney.

When the boiler is being forced at above its normal power the temperature in the chimney will be higher than in ordinary circumstances, and it will therefore be more powerful. The maximum temperature which should exist in the chimney of a heating installation at any time should not exceed 550° F. In ordinary normal working the temperature in a chimney should not be more than from 150° to 200° above the temperature of the water in the boiler. Thus, when the boiler temperature is 180° the chimney should be sufficiently powerful to draw the required quantity of air through the furnace when its temperature is 330° to 350°.

Theoretical Formula for Size of Chimney.

Assume W is the total B.T.U. which the boiler is required to give off.
$x'' \times y'' =$ size of chimney.
H feet $=$ height of chimney.

The amount of fuel burnt per hour $= \dfrac{W}{8{,}000}$

The volume of flue gases $= \dfrac{W \times 300}{8{,}000}$ cubic feet per hour.

Velocity of flue gases up chimney =

$$\frac{W \times 300 \times 144}{8{,}000 \times 3{,}600 \times xy} \text{ ft. per sec.} = \frac{W}{666\,xy}$$

Loss of head when gases pass up a flue $x'' \times y''$ at a velocity $\dfrac{W}{666\,xy}$ feet per second

$$= \rho\,\frac{2\,(x+y)\,H \times 144}{12 \times xy} \times \frac{\left(\dfrac{W}{666\,xy}\right)^2}{2g}$$

$$= \rho\,\frac{(x+y)\,H\,W^2}{x^3\,y^3} \times \frac{1}{1{,}190{,}200} \text{ feet of flue gases.}$$

The ordinary value of ρ may be taken for present purposes as .007.

Assume the pressure required for forcing air through the boiler and flues as far as the entrance to the chimney is 0.2″ of water pressure.

Since air at 350° is $\frac{1}{1,267}$ times the density of water at 62°, we have 0.1″ water pressure

$$= \frac{0.1 \times 1,267}{12} = 10.5 \text{ feet of flue gases.}$$

The formula, therefore, for the minimum size of chimney is, on the above assumptions

$$\frac{H}{D_i}\left(\frac{D_o}{1 + a\,t_o} - \frac{D_i}{1 + a\,t_i}\right) = 21 + \rho\,\frac{(x + y)\,H\,W^2}{x^3\,y^3} \times \frac{1}{1,190,200}$$

Substituting the value 0.076 for D_o and D_i, and $\frac{1}{461}$ for a, 60° for t_o, and 350° for t_i and .007 for ρ.

$$H \times 0.318 = 21 + \frac{(x + y)\,H\,W^2}{x^3\,y^3 \times 17 \times 10^7}$$

Approximate Formulæ.

According to Reiche the ratio between the area of the chimney and the area of the grate should be 1 : 4 for hard coal and 1 : 6 for brown coal.

The same authority gives the rule that the height of the chimney ought to be at least 25 times the diameter, with a minimum of 50 feet in all cases. This, however, appears to be a counsel of perfection. Many chimneys of less than half this height are perfectly successful.

The following table has been compiled to give the approximate size of chimneys for different duties. These figures must be accepted as considerably larger than are strictly necessary to do the duty.

The following table gives the approximate number of pounds of coal which can be burnt per square foot of chimney section when the ratio between the grate area and the chimney section is 8/1.

Height of Chimney in feet.	Pounds of Coal per hour per sq. ft. of Chimney.	Velocity of flue gases in ft. per sec. up chimney when each pound of coal gives off 300 cubic feet of flue gases.
10	60	5
30	76	6.3
40	93	7.75
50	105	8.75
60	116	9.65
70	126	10.5
80	135	11.3
90	144	12
100	152	12.6
110	160	13.3

The diagram, Fig. 110, shows a range of practical sizes suitable for use with ordinary heating installations.

Regulation of the Combustion.

For any given thickness of fire and quality and size of fuel the intensity of the combustion is determined absolutely by the amount of air which passes through the fire.

With the same quantity of air passing through the fire, a difference in the quality and size of the fuel causes a difference in the rate of combustion, and therefore an alteration in the temperature of the furnace. But with fuel of constant size and quality the amount of the combustion is regulated entirely by the quantity of air which flows through. This quantity of air is itself determined by the difference in pressure between the inside of the grate and the interior of the furnace. A slight difference in the absolute pressure under the grate only makes an insignificant difference in the rate of combustion unless it reduces this difference.

In order, therefore, to alter the rate of combustion, an alteration must be produced in this difference of pressure.

Methods of Regulating the Combustion.

It is the common method of regulating the combustion to control the area of the inlet through which the air finds its way under the grate. The velocity with which the air flows through the fire is not altered until the reduction of the area of the inlet so alters the proportions of pressure in the system of flues as to produce a difference in the *relative pressures above and below the fire.*

Even then the quantity of air only varies as the square root of the difference of pressure. This is the reason for the commonly observed, but rarely understood, fact that frequently little change in the temperature of a furnace or reduction in fuel consumption is brought about by even a large reduction of inlet area. The first effect of a reduction of the inlet area is to increase the difference in pressure between the outside air and the under side of the grate, but if, as usual, the resistance of the fire itself is only a small part of the total resistance, the effect of this partial closure is merely to produce a general reduction of pressure throughout the entire system of flues, and not to alter to any great extent the difference in pressure below and above the fire grate.

The control of the quantity of air passing through the fire is effected

(1) By regulating the resistance of the air passage by reducing mechanically the area of the inlet or outlet through which the air or flue gases have to find their way, or

(2) To some extent by controlling the total intensity of the draught drawing the air through the furnace.

It is not easy to make a damper both free in working and airtight if it is fixed in the flue gases. In such a position it is liable to collect soot and dirt. Hence the method generally adopted where it is necessary for the damper to work freely (as in all automatic systems of regulation) is to place the damper on the inlet quite outside the boiler where no flue gases reach it.

It has already been explained that the existence of a constant circulating pressure in the chimney tends to limit the effectiveness of control when this is accomplished by restricting the inlet air passage.

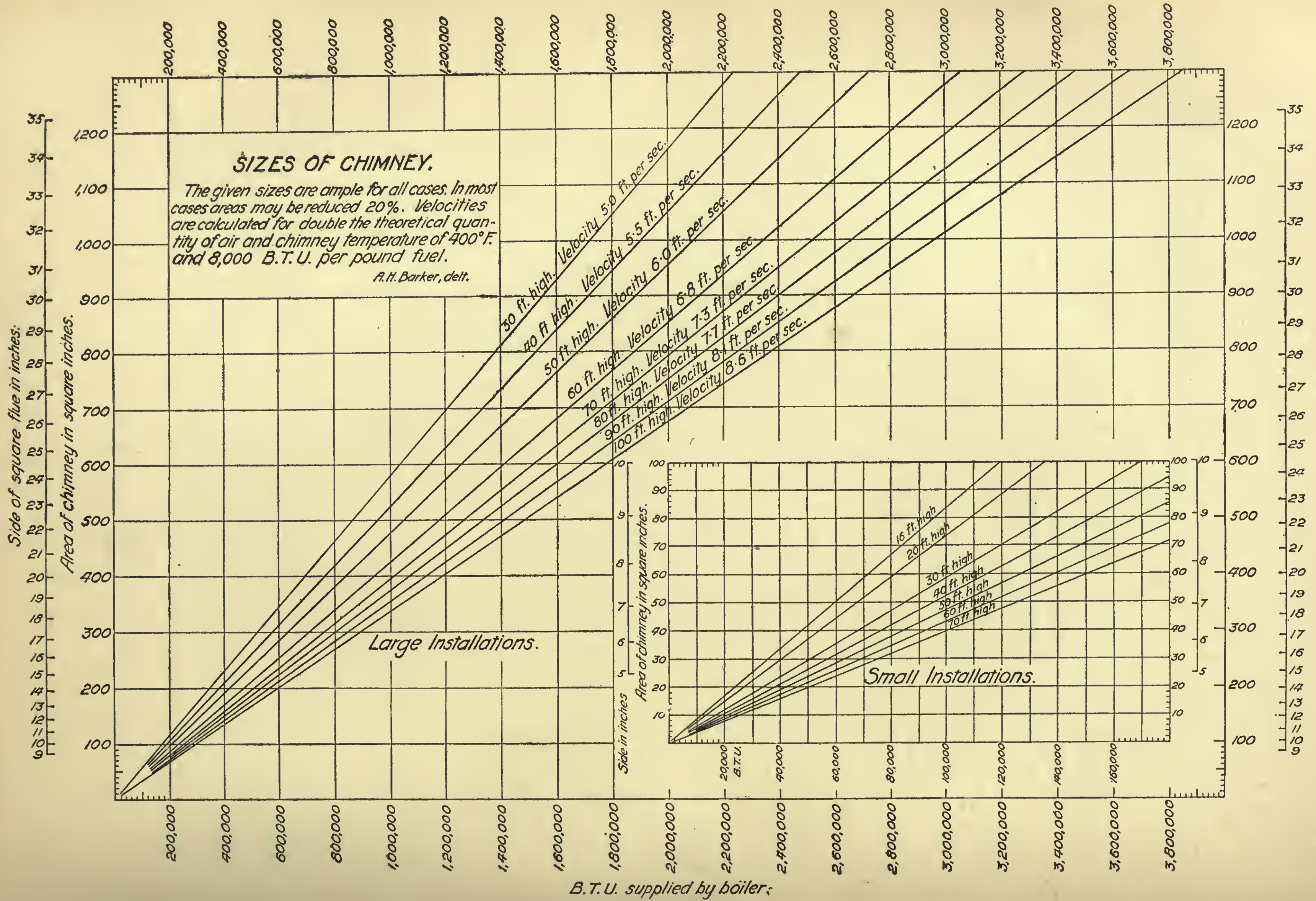
In order that regulation by this means may be at all effective it is therefore necessary that the base of the boiler should be airtight, so that no air can reach the under side of the grate otherwise than through the authorised passage, namely, the draught regulation door. The draught regulation door, too, should be very carefully fitted so that when it is closed the entire space underneath the fire bars is as nearly airtight as practicable. Only in this way can any considerable control of the fire be obtained where the chimney is powerful.

A secondary method of regulation, used only in combination with method 1, is to control the intensity of the draught or motive force by admitting cold air in varying quantity into the chimney so as to take the "sting" out of the draught by cooling the gases in the interior of the chimney. This diminishes the suction power, which the high temperature and specific levity of hot gases give them. This method is not used alone as the degree of control it gives is not in ordinary cases sufficient.

If the base of the boiler is absolutely tight, a secondary arrangement of this kind should not be *necessary,* although in practice it is very difficult to ensure that the base of any boiler shall be so tight that no such secondary regulation is desirable.

The regulation of a boiler must in any case be a somewhat gradual and slow process, where it is effected only by controlling the quantity of air. For instance, assume that the incandescent fuel and furnace gases are at a given moment communicating to the water in a boiler, say, 300,000 B.T.U. per hour, and that the boiler is of such a size that in order to obtain this output the fuel must be maintained at a bright red heat. If the boiler is of the usual type, designed for a fairly long run, there would probably, in these circumstances, be in the furnace a mass of, say, 6 cubic feet of incandescent fuel, say coke, weighing about 28 lbs. per cubic foot, and having a specific heat of 0.2 at a temperature of, say, 2.300 deg.

Assume that at this moment it is desired to reduce the output to 200,000 B.T.U. per hour, and that in order to effect the reduction the air supply to the furnace is for the time entirely shut off. The total amount of heat



SIZES OF CHIMNEY.

The given sizes are ample for all cases. In most cases areas may be reduced 20%. Velocities are calculated for double the theoretical quantity of air and chimney temperature of 400° F. and 8,000 B.T.U. per pound fuel.

A.H. Barker, delt.

FIG. 110.

calculated above 60° contained in the 6 cubic feet of coke, which would weigh about 6 × 28 = 168 pounds, would be

$$28 \times 6 \times (2{,}300^\circ - 60^\circ) \times 0.2 = 75{,}000 \text{ B.T.U.}$$

In order to cool this incandescent mass down to dull red heat (say, 1,800° F.) which would probably correspond to a total heat emission of 200,000 B.T.U. per hour, it would be necessary to extract from the fuel about

$$0.2 \times 168 \times (2{,}300^\circ - 1{,}800^\circ) = 16{,}800 \text{ B.T.U.}$$

Even without consuming another particle of fuel, therefore, this amount of heat would be supplied by the incandescent fuel to the boiler. It would maintain the mean heat supply at the same rate for about

$$\frac{16{,}800}{300{,}000} \times 60 = 3.4 \text{ minutes,}$$

or at a uniformly reduced rate for, say, 7 minutes, the stored energy, of course, being derived from the chemical energy of the coke which had been previously burnt.

It is therefore clear that in the case of a boiler with a large firepot capacity, rapid or prompt regulation is not possible if all the fuel in the firepot is allowed to become incandescent at the same time. If prompt regulation is required only a small amount of the fuel must be incandescent at one time, as in some forms of magazine boilers when the flue gases leave the furnace through an opening near the level of the fire grate.

Mechanical Means Employed.

The automatic regulation of the fuel consumption in a boiler is effected by means of mechanism which automatically opens or closes an inlet draught door, according to the temperature or pressure, as the case may be, of the water or steam in the interior of the boiler.

In the case of water the motive power for operating the dampers is derived in almost all cases from the expansion of some substance, solid, or liquid, or gaseous. By means of a suitable system of levers very minute amounts of expansion of this substance can be so magnified that even an alteration of one or two degrees in the temperature of the water in the boiler will move the draught door appreciably.

In the case of a steam boiler the same end is effected by means of the alteration in pressure expanding or contracting a chamber or space having flexible bounding surfaces. When the pressure increases this variable chamber is extended in size, and a simple arrangement of levers transmits the expansion to the draught door, which is partly closed for every increase of pressure and vice versa.

In other cases, in a steam boiler, the pressure is made to maintain a column of liquid (generally either water or mercury) of a certain height.

A reduction in the pressure causes a reduction of the height of the column and vice versa. The movement of the top of the column is transferred to the draught regulating door by a float floating on the surface of the liquid column, or by causing the surface of the liquid itself to regulate or close the air passage.

Whatever the mechanical means employed, the conditions of satisfactory regulation are—

(*a*) That the entire mechanism must work with a minimum of friction.

(*b*) That the gear is so designed as to be fully open at the lower limit of temperature and pressure and entirely closed at the upper limit.

(*a*) This result can be secured by proper attention to the mechanical conditions in which the mechanism is placed.

Such devices as chains passing over pulleys which are commonly employed, should be eschewed, for the reason that they give rise to an unnecessary amount of friction, and if not properly attended to are apt to stick quite tight owing to an accumulation of rust or dirt.

It is always possible to arrange freely working levers in place of pulleys, which attain the same end in a condition much freer from friction than can be secured with pulleys.

It is the common practice to transmit the movement by means of chains, and if these are employed, the pull of the chain should always be in the same plane as the movement of the levers, so that there is no side friction caused by the oblique arrangement of forces.

It is the object of the regulator to keep the temperature or steam pressure, as the case may be, from rising above a certain limit, or from falling below a certain limit. With an ideal system of regulation the upper and the lower limits would be fairly close together. The closer together they can be kept the better the regulation of the boiler would be. In practice, however, the reservoir of heat in the mass of incandescent fuel commonly found in a boiler prevents the limits from being very close together.

If an attempt is made to keep them too close together it introduces the danger of what may be called "hunting" of the regulator, especially when some little friction exists in the mechanism.

Hunting of Regulator.

The phenomenon we have called "hunting" is brought about as follows :—

Assume that at a certain moment the fire is dull and the temperature of the water or the pressure of steam relatively low. The draught door is then held wide open by the regulator, the temperature or pressure being near the lower limit. The vigorous draught so produced will cause the fire to burn up brightly, and to attain a high temperature within the body of the fire, especially near the middle or core. Before the effect of the increased combustion can be transferred to the water, that is, during the time the body of the fire is becoming hotter, it gradually accumulates a reservoir of heat, and a temperature which, when normal conditions of conduction are established, may be more than sufficient to carry the heat of the water or steam pressure considerably beyond the desired point. The temperature of the water or the pressure of the steam *lags* behind the temperature of the fire, owing to what may be described as the heat inertia of the fire and water. Especially is this the case in a water system, where each degree difference of water temperature may mean the absorption of many thousands of B.T.U.

The temperature in the body of the water only slowly rises however perfect the mechanism may be. The damper only shuts completely when the temperature of *the water* has risen to the desired point. By this time there is sufficient heat in the body of the fire to raise and maintain the temperature of the water considerably beyond that point.

During the period of elevated temperature or pressure the damper remains shut, and the temperature of the fire begins to be reduced. It may continue to fall until it is lower than is required to maintain the heat. But the temperature *of the water* does not *immediately* fall. It requires some minutes at least after the fire temperature has fallen too low before the temperature of the water has dropped to such a point that the damper opens at all, and although the damper is then opened the temperature in the water continues to fall until the damper is open very wide once more, and the whole process begins again.

The effect is that the temperature or pressure oscillates between wide limits—in many cases far beyond those at which the damper is fully open or completely closed. This is the phenomenon which may be called "hunting" by analogy with the similar phenomenon produced by an improperly loaded and adjusted steam engine governor.

This bad effect is obviously intensified by any friction that may exist in the mechanism, for in this case the damper does not shut gradually, but is closed in one movement when the temperature has already risen considerably too high, or is opened wide at one movement when the temperature has fallen considerably too low, thus intensifying the hunting action. It is also intensified if the boiler is more powerful than it need be.

(*b*) The other condition of first rate importance is that the base of the boiler should be as airtight as it is possible to make it, so that no air can find its way to the under side of the grate otherwise than by the authorised channel, namely, the draught regulating door. If it is not so the fire may, when the draught is keen, get enough air to exceed the desired pressure, even when the draught door is completely closed. As already mentioned, this effect may be reduced by admitting air into the chimney, thus controlling the intensity of the draught simultaneously with the partial closing of the inlet door.

If these two conditions are perfectly fulfilled, given satisfactory mechanism, this is the most that can be effected in the way of regulation by controlling the fire only.

But the control of the amount of fuel burnt is not the only or even the most perfect method by which the control of *the output* of heat can be obtained.

Another method, less generally adopted and very little used, by which considerable accuracy can be obtained, consists in admitting more or less air above the furnace, so as to cool more or less the products of combustion after leaving the furnace. By this method an almost instantaneous regulation can be obtained, though at the cost of wasting the fuel. Any admission of cold air into the flues of the boiler necessarily results in the wasting of the fuel.

The inlet of air can be controlled in a similar manner to that already described.

Practical Considerations.

There are many practical matters to be considered in relation to chimneys of heating installations, which though they have little to do with the theory of the matter, may have an important bearing on the success of a boiler plant.

An inefficient chimney is probably responsible for more boiler troubles than any other single cause.

In the chapter on boilers a number of causes of boiler trouble have been enumerated. In case the trouble is definitely proved to be due to the chimney, the following method of procedure will generally enable its cause to be located.

Method of Applying Draught Gauge.

The ordinary method of applying a draught gauge to a chimney conveys very little information. Boilers of different designs require substantially different air pressures in order to force the necessary air through the furnace and flues, and a chimney which would be quite adequate for a boiler of one design might be entirely inadequate for boilers in which the resistance is greater.

It has been shown, too, that the resistance varies very much according to the quality of the fuel employed, and according to the condition of the fire. Merely to apply a draught gauge, therefore, to a chimney in one spot gives very little information. Many persons apply a draught gauge without any very clear idea of what they intend to ascertain by means of its readings.

If a chimney of suitable height and apparently suitable size is insufficient to produce certain results, the cause may be one of several.

In applying a draught gauge it must always be clearly borne in mind that the total circulating pressure generated by a chimney depends solely on the height and the temperature inside and outside. The generation of this total pressure when the proper temperatures exist may be regarded as a certainty, since it follows directly from an invariable natural law. This total draught is always divided among the five or six sections enumerated on page 417. If the draught does not produce the expected result it is necessary to ascertain why. The reason may be

(1) That there is some counteracting influence at work, such as air leakages, wind pressures, or other cause, alluded to in another part of this chapter.

(2) That some of the resistances are excessive.

In the latter case it is necessary to ascertain which of the resistances is excessive. The only method by which this can be done is to apply a micrometer draught gauge having two ends connected at different points of the flue.

The ordinary draught gauge consisting of a U tube made of glass is not sufficiently accurate for practical work.

The most generally useful points of connection are at the outlet into the chimney and underneath the fire grate of the boiler.

The reading of a draught gauge so fixed shows the amount of air pressure which is being employed in forcing the air through the fire grate, the fire, and flues of the boiler.

If the gauge is connected between the ashpit and the open air, it gives the pressure employed in generating the velocity through the inlet. If it is connected between the ashpit and the furnace, the result shows the loss of heat in forcing the air through the fire only.

A gauge when fixed between the base of the chimney and the open air gives the sum of the resistances of the inlet, grate, fire, and flues of the boiler. The difference between these and the calculated power of the chimney gives the resistance of the shaft.

Since the resistance of the inlet varies according to the degree of opening of the draught door, such indications are generally useless.

The effect of an excessive resistance in the chimney is to cause an excessive loss of head between the base of the chimney and the top, and a consequent reduction in the circulation pressure drawing air through the boiler. If the reading in the water gauge indicated at the base is small, it may mean either that there is an excessive resistance in the chimney or very little resistance in the boiler and flues.

A very low draught reading at the base of the chimney is not at all inconsistent with a large excess of air passing through the boiler. Any complete information in regard to a boiler should include what draught is necessary in different conditions to draw the requisite quantity of air through the boiler and fire, otherwise the engineer has to rely on his own experience only. If he has no specific information on this point, it is almost useless to apply a draught gauge at all.

The air pressure employed in driving the air through the draught door and conversely the approximate quantity of air per hour, due to an observed draught, may be calculated as follows :—

As has been explained, the partial closing of a draught door causes a partial vacuum under the grate. There is, of course, a greater vacuum at the base of the chimney.

If then 6,000 cubic feet per hour pass through the draught door, and it has an area 8″ × 3″, the velocity generated must be calculated as follows :—

The effective area of the inlet is $\frac{8'' \times 3'' \times 61\%}{144} = 0.102$ sq. ft.

Velocity in feet per second $\frac{6{,}000}{3{,}600 \times 0.102} = 16.3$ ft. per sec.

Head of air column $h = \frac{v^2}{2g} = 4.1$ feet.

Water gauge $\frac{4.1 \times 12}{800} = .06''$.

Note must be taken of the fact that the whole area over which this velocity exists is not the full area of the inlet, but about 61% of it, owing to the phenomenon known as the "vena contracta" (see chapter on "Flow of Fluids").

To calculate the quantity of air flowing through a boiler given an accurate measurement of the vacuum under the grate, this calculation may be reversed.

It is necessary to ensure that the total draught in the chimney is equal to the calculated value. It should be clearly borne in mind that if a chimney is sound, the total circulating pressure generated by it depends essentially and solely on its height, and the temperatures inside and outside. Given the temperatures, the production of the circulating pressure is a direct result of an invariable law of nature.

In order to ensure that the internal and external temperatures are as calculated, they should be measured by means of thermometers fixed in the flue gases and open air respectively.

If the internal temperature is not as high as it ought to be, the reason should be investigated. The reason may be one of several, of which the most general is inleakage of air.

A single small leakage into a chimney will admit a surprisingly large quantity of air, and a number of leakages, such as are frequently found in old chimneys, is quite sufficient to reduce the natural draught up a chimney by one half or more.

The existence of such leakages can be generally detected by observing the difference of temperature between the flue gases at the base of the chimney where they leave the boiler and the outlet at the top. If this difference is considerable, the existence of air leakage can be at once suspected. The general difference between the temperature at the base of a tight chimney and at the top is not more than 10° or 20°.

Leakages can generally be detected by the flame of a candle or match, which when held near the leakage will be drawn into the interstices between the bricks or joints. Such leakages are generally to be found around or through soot cleaning doors, or through the brickwork setting (if any exists) of the boiler, or between the flue connections from the boiler to the chimney, especially where these are made of metal.

To stop these leakages old chimneys have sometimes to be pointed throughout, as leakages exist everywhere between the bricks.

Another effect which is frequently responsible for defective draught in a chimney is the existence of damp or wet within the chimney. Such damp immediately reduces the temperature of the flue gases to a very marked extent. If this condition is found to exist, steps should be taken to ascertain the cause and to remedy it. It frequently results from leakage of the brickwork in a rain storm, and is often accompanied by leakage of air.

Sometimes also it results from inefficient drainage of the house, roofs, the blocking up of gutters, etc.

Down-Draughts in Chimneys.

A difficulty is sometimes experienced in first starting a boiler fire, and the same applies to open fires. There will sometimes be very bad down draughts in a chimney flue for some time after the fire is first started, resulting in the blowing of enormous volumes of thick smoke into the boiler chamber in which the fire is placed. This may be due to the dynamic action of the wind outside, which is dealt with below. But it may also be due to the chimney being very cold to begin with.

If the chimney is placed in an outside wall of a high building, the interior of it will have about the same temperature as the outside air when the fire is first lighted. If at the same time the other parts of the house are heated, and especially if a length of horizontal flue intervenes between the fire and vertical part of the stack, the house itself as a whole, or the staircase, will act as an upcast flue, and will actually draw air down the cold chimney so long as the temperature in the house is higher than in the chimney.

When the fire is first lighted this down draught carries all the smoke and products of combustion into the room in which the fire is placed, and this will continue more or less intermittently until the temperature of the chimney has been raised so high that the current is reversed. Until this state is arrived at, however, the smoke pours in large volumes into the room.

This cannot be stopped by any application of patent chimney pots, which is the general method attempted by the uninitiated to remedy matters. One might as well put in a patent chimney pot to stop the flow of a stream of water down a hill.

Perhaps the easiest way of stopping it is to fix one or two gas jets within the chimney about half way up, guarded by an airtight access door. These can be lighted for a few minutes previous to the fire being lighted. The existing down draught will carry the hot gases from the flames downwards into the chimney and will suffice to warm it sufficiently to give the requisite upcast. After the fire is lighted the heat from the fire will maintain the chimney temperature continuously. The gas jets can then be turned out.

The Effect of Wind.

Difficulties are sometimes caused by wind, which may cause intermittent down draughts in the chimney, which are very annoying and even dangerous. This effect is generally caused by the upper outlet from the chimney being arranged in such a position that it comes within the area of dynamic pressure generated by the wind acting on some adjacent wall surface.

Thus, if the outlet from a boiler chimney is placed near to a wall, the outlet being lower than the top of the wall, if wind drives up against the wall it generates considerable pressure, which may easily be greater than the total circulating pressure set up by the hot gases in the chimney. When this is the case, the excess of pressure drives the smoke down the chimney and drives it out into the boiler chamber.

It has been explained on page 186 what pressure is set up by wind at varying velocities, and it is easy to calculate what velocity of wind would neutralise the action of any given chimney placed in such a position.

The remedy for this trouble is either to raise the outlet of the chimney above the level of the top of the wall, or above the ridge of the adjacent roof, up to such a level that the wind has a clear course past the top of the chimney.

Another remedy is to form a flue, if possible, larger than the chimney itself down into the boiler chamber from the neighbourhood of the smoke outlet. This will equalise the pressure between the chamber and the chimney outlet, and the whole circulating pressure of the chimney will then be available for producing the draught.

Troubles of this kind can sometimes be remedied by the provision of openings into the heater chamber from an area or from some other quarter such that similar dynamic pressures tend to be produced in the heater chamber as are produced in the neighbourhood of the outlet, thus counteracting the effect of the external dynamic pressure.

A similar effect, but due to a wholly different cause, may be produced if the wind is so deflected by surrounding wall and roof surfaces as actually to blow down the chimney. In the latter case, but only in this case, the trouble may be cured by a well designed cowl, which guides these currents of air so as to aid instead of hindering the draught. These are the only circumstances in which a cowl or patent chimney pot can conceivably be of the slightest use.

A cowl is worse than useless when the down draught is due to pressure only, arising from either of the previously explained causes. In any case, a cowl must be of such design that it cannot impede the flow of air when the chimney is working in its normal way.

Resistances in the Flues.

In investigating the sufficiency of a chimney, as in all other investigations connected with a boiler of a heating installation, it is in the first place necessary to know what is the number of B.T.U. which the boiler is expected to or ought to develop.

The general theory of chimneys has been explained, and it has been shown what is the effect of excessive resistance in various parts of the system of flues.

The proper application of a suitable draught gauge, combined with knowledge of the pressure required to circulate sufficient air through the boiler, enables the existence of chimney troubles to be definitely located. Even without previous knowledge of the boiler resistance, accurate observations of a draught gauge, combined with knowledge of the fuel used and observations of the consumption of fuel burnt per hour, will enable a good deal of information in regard to the resistance of the chimney to be obtained on the principles already explained.

The causes which may produce excessive resistance in the chimney flue are

(1) That the chimney is not on the whole large enough.
(2) That it is constricted at some point.
(3) That there is an accumulation of soot or dirt somewhere within the chimney.

Chimneys are frequently rendered inefficient by the fixing of a chimney pot which is too small, owing to the eccentric ideas on the subject of draught often entertained by builders and others who ought to know better.

The foolish notion, which is quite prevalent, that a small chimney pot or outlet is conducive to an efficient chimney, quite accounts for many of the chimney troubles which are sometimes experienced.

In many cases, also, chimneys are rendered inefficient by the use of one of the many types of patent chimney pots, some of these being ingenious arrangements of passages and baffle plates which appear to be intended to prevent down draught, but which have in fact the usual result of stopping the normal flow of gases which might be produced by the heat.

These appliances should be regarded by the heating engineer with great suspicion. Indeed, one of the first points to look for in the investigation of an inefficient chimney is whether or not one of the patent draught improvers have been put on.

The next thing is to investigate whether there is some unnoticed interior constriction in the chimney. These are frequently brought to light in old houses, in which either the chimney has been encroached upon by some alteration of bygone days, or in which the original builder has entertained the same eccentric notions on the subject of draught as his modern brother sometimes does. Sometimes chimneys are carried up in a wall of a certain size up to a certain height, above this the flue is smaller. If the size has been measured in the lower part the engineer assumes that the flue is of the same size all the way up.

Sometimes the draught may be a good deal checked by sudden sharp bends in the flue. These bends in themselves may not greatly increase the resistance, but they are apt to form lodging places for accumulations of soot.

There is a prevalent notion that the horizontal portion of a flue constitutes a danger to the draught, and rules of thumb are given according to which the horizontal portion of a flue must not be more than a certain proportion of the vertical length. Such rules are based on a fallacy. The only disadvantages attendant on a horizontal portion in a flue are (1) the mere increase of length causing increase of friction. (2) The danger of accumulations of soot which are difficult to remove, especially if the flue is underground. (3) The loss of heat in passing through the flue. (4) The difficulty of cleaning and of keeping the cleaning doors airtight and watertight, and the consequent leakage of water and air into the flue, causing chilling of the gases, especially if the flue is underground. (5) The difficulty of starting the draught, for the heat has to travel the whole length of the horizontal portion before it reaches the vertical, which therefore does not begin to draw until after the horizontal portion is made hot. During the process the absence of draught causes smoke troubles.

CHAPTER XXI.

ON THE HEATING AND HUMIDIFYING OF AIR FOR VENTILATION.

Heating.

AIR for ventilating a room or hall is commonly warmed by passing it over a battery of heating surface supplied with hot water, steam, or hot flue gases, as the case may be. By far the most common medium for this purpose is low pressure steam.

The amount of heat which 1 square foot of heating surface will give up to air passing over it, varies widely (1) according to the character of the surface and (2) according to the velocity of the air.

The calculations consist simply in the determination and application of the correct value of K, the co-efficient of transmission for the particular type of surface adopted. Such surfaces vary widely in character. It is not possible to obtain accurate values otherwise than by specially careful experiments on the particular type of surface adopted.

For general purposes it is only possible to give the value for one or two of the most ordinary types of heating surface.

Up to within the past few years the type of surface most commonly adopted for warming air for ventilation consisted of a battery of 1″ pipes fixed close together, having a space of about ¼ or 5/16ths of an inch between them.

The following values for K for this particular case are given by Rietschel, reduced to English measure, and these values are plotted on Fig. 111, from which the large increase in the value of K with the velocity will be appreciated.

It is probable that a proportionate increase would be found if other types of heating surface had been investigated with the same care. If, therefore, the value of the co-efficient is determined for one particular velocity of the air, it will probably be safe to assume that a proportionate increase in this value is produced by increasing the velocity of the air over the surface.

The value of K is about 30% greater when the air impinges vertically on the heating surface than when it passes over it in a direction parallel to its surface. In cases where the type of heating surface is one of which accurate figures are not obtainable, it is advisable to be well on the right side in providing for it, or at any rate to make such provision that the additional heating surface can be added if it is found to be necessary. This can generally be done without difficulty.

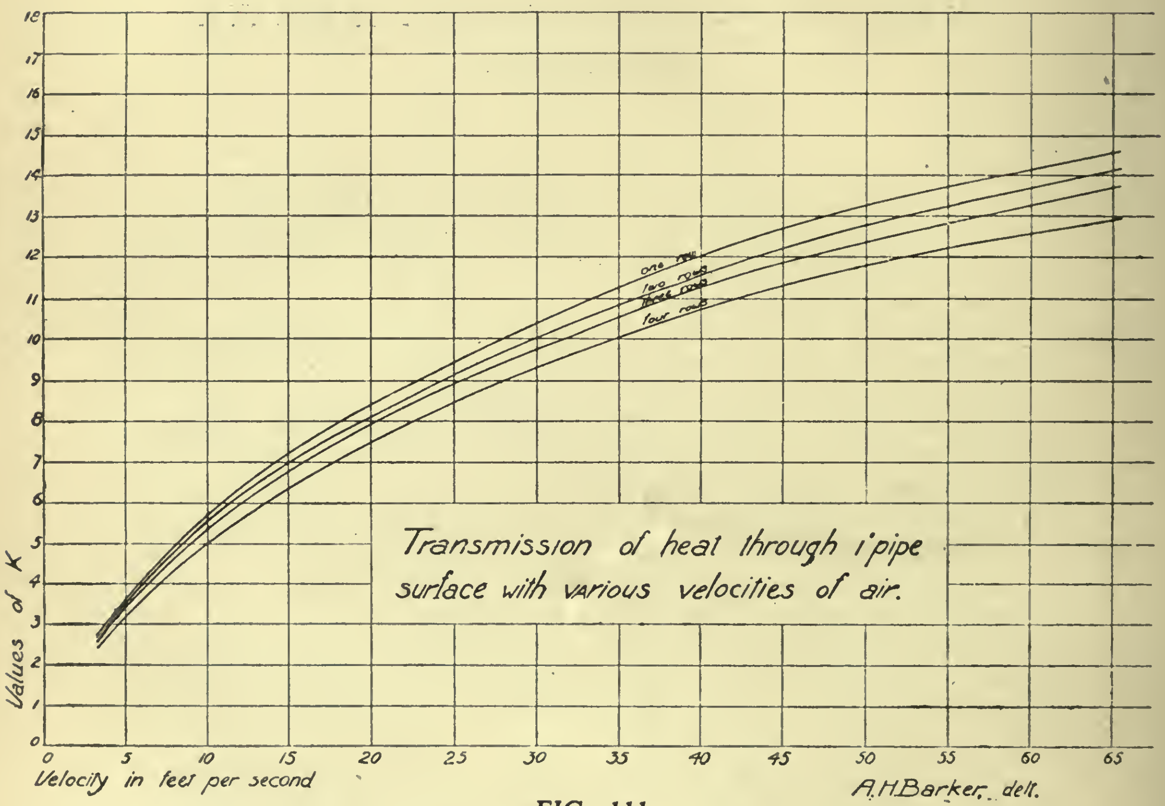


FIG. 111.

Table XXVII. shows Rietschel's values for K where the heat is supplied by flue gases. These are also plotted in Fig. 108*a* on page 416.

In this case there is normally a great difference between the inflowing temperature and the outflowing temperature of the flue gases. The equations on page 288 are applicable to such a case as this.

In the case of a steam battery the temperature of the heating surface is almost uniform, more or less, over the whole surface. This is not, however, the case with flue gases, which are commonly reduced very much in temperature before they are ejected.

The different arrangements of heating surfaces are so numerous that it is impossible to give figures for even a very small part of them, and the value of the co-efficient to be adopted must be left to be determined by experience in cases where no accurate experimental information is available.

Calculation of Heating Surface.

The calculations required for determining the amount of heating surface for a given duty are as follows :—

Let V be the volume of air required to be heated from t_1 to t_2 per hour.

It is first necessary to ascertain the weight of this air (W) which will be found from the formula explained in Chapter V., namely :—

$$W = V \times \frac{461}{461 + t_1} \times 0.0863 \text{ pounds.}$$

The quantity of heat (H) required for heating it is

$$H = W \times (t_1 - t_2) \times 0.238$$

The velocity of the air over the heating surface is calculated as follows :—

Assume that the centres of the pipes are fixed S'' apart and that the projected area of the heating battery normal to the direction of flow is A square feet. Then every foot run of pipe in the front section of the battery is accompanied by a space one foot long by $(S - D)''$ wide, where D is the outside diameter of the pipe employed.

The total length of pipe (l) which can be fixed in a space of A square feet is

$$l = \frac{A}{\frac{S}{12}} = \frac{12\,A}{S} \text{ feet run.}$$

Therefore the total free area (a) for the air is

$$a = \frac{12\,A}{S} \times \frac{(S - D)}{12} = \frac{A\,(S - D)}{S} \text{ sq. ft.}$$

The linear velocity (v) of the air per second, therefore, where V cubic feet per hour pass through this first section of the battery is

$$v = \frac{V}{3{,}600} \div a$$

This expression will give the required velocity and the corresponding value of K can be taken from the diagram.

This value of A may, of course, only be provisional, and in case the latter portion of the calculation shows the value for A first selected unsuitable, A can be increased as may be necessary.

We have then the total surface necessary (F) would be found from the following equation, where H is the total quantity of heat obtained from the previous calculation :—

$$H = F \times K \times \left(t_3 - \frac{t_1 + t_2}{2}\right)$$

This may be reduced to lineal feet of pipe by the following equation :—

$$F = \frac{D \times \pi}{12} \times L$$

or

$$L = \frac{12\ F}{D\ \pi}$$

Calculations similar in principle may be made for any other form of heating surface which it is desired to employ.

Humidifying of Air for Ventilation.

It has been already explained that when air is heated it acquires a largely increased power of taking up moisture. The effect is not, of course, to make it dryer in the absolute sense of removing moisture from it, but only in the sense of removing it further from the dew point, that is, to increase its relative dryness.

The lower the relative humidity the more rapid is the evaporation from every substance with which the air comes into contact containing water in its composition.

The belief is commonly held that stoves or other heating surface at a high temperature make the air drier than do radiators or heating surfaces at lower temperatures. This, however, is not strictly correct. It is merely the rise of temperature of the air which causes a diminution in the relative humidity. The means whereby the temperature is attained has no different effect, whatever the means may be.

There is no doubt, however, as has been previously explained more in detail, that the use of high temperature heating surface produces an effect on the air which may be called "burning," and which probably consists in the charring or carbonising of many of the organic particles which are found in common house dust. It is probable, at any rate, partly, the smell and the mechanical effect of the charring process which produces the effect known as "burning of air."

It is possible, too, that the greater variation of temperature at different levels produced by excessively hot radiating surface or the introduction of very hot air has something to do with the production of the well-known unpleasant effect. There is certainly no difference between the actual dryness of the air whatever the temperature of the heating surface may be.

Humidity of the Outside Air.

The leading physical facts relating to moisture in air have been explained in Chapter IV., including the means for measuring the humidity. The condition of the outside air as to dryness is variable. Relatively large variations are found in this respect, not only between the different seasons of the year, but also from day to day and from hour to hour. It is therefore one of the most difficult problems in ventilation to take accurate account of. At any season of the year the relative humidity may vary from a low value almost to complete saturation.

The beautiful diagrams, on page 57, published by the Meteorological Office give the *mean* humidity of the atmosphere at two stations in this climate for all hours of the day and night during the various months of the year. These diagrams, of course, take no account of the maximum diurnal variations. They show, however, very clearly how the mean humidity taken over a period of 20 years varies at different seasons of the year at all hours of the day and night. A comparison of the two gives a very good idea of the relation between the climates at the two places for which the diagrams are drawn.

It will be seen that in certain winter months the mean humidity of the atmosphere is very high, which implies that the air is at those times often almost saturated.

The very lowest degree of humidity in the outside air ever recorded in England is 23%; 85% and over is considered damp, 65% moderately dry, 50% dry, 35% very dry.

The physiological effect of warming air is to increase the rapidity with which heat and moisture are lost from the bodies of persons, especially from those parts such as the nasal passages and throat, which are normally kept moist or wet by the body mechanism, and to produce in consequence a feeling of dryness and discomfort. This effect diminishes the power of the body mechanism to resist the attacks of disease germs which are normally prevented by these moist surfaces from entering the body.

It appears desirable, therefore, for health and comfort to provide means for humidifying the air in connection with every high class system of ventilation. It must, however, be stated that there are a few authorities who hold the view that provided air is not extremely moist its degree of dryness is not of very considerable importance so far as its healthiness is concerned, and that only very few persons are affected by dry air in a sense prejudicial to comfort.

These authorities conclude that the quantity of moisture in the air is a wholly unimportant matter, that there is no justification for the spending of money to maintain any considerable degree of humidity in healthy air. It is, of course, a matter of hygiene and not of engineering to decide on such a point as this.

It is certainly a fact, as will be seen from the calculations below, that in order adequately to moisten air, the absorption of a large amount of heat is, in fact, necessary and entails a considerable annual expense and a good deal of troublesome attention.

It is the experience of the present writer that a certain large part of the dislike of central heating systems, which is so common in this country, is removed if the air is humidified as well as warmed.

It must, of course, be remembered that every human being gives off with his breath and from the surface of his body a considerable quantity of moisture, and the effect of this on the relative humidity of the air supplied per head for ventilation must not be lost sight of. It certainly operates to increase the humidity or dryness to a degree which will be discussed later.

Desirable Degree of Humidity.

The exact degree of humidity which is most favourable from a hygienic point of view for ordinary air is not accurately known. It is probable that, provided the air is not too dry or too wet, the exact degree of humidity is not of considerable importance within very wide limits, say, from 40% to 70%.

The most desirable hygrometric condition, all things considered, for the interior of a living-room or meeting hall appears to be about 50% to 60% of saturation. Anything up to 75% is not considered harmful, except for the fact that moisture contained in vitiated air probably emanates from human beings. Just as in the case of the CO_2 impurity, though the moisture in itself may not be deleterious, yet the organic smells which accompany it may be very objectionable, if not on that account positively harmful.

The truth appears to be that dry air is less harmful than air which is too moist, but that air at a medium moisture, such as 50% or 60%, if not in the very best condition for breathing, is at least not distinctly more harmful than at any other degree of saturation.

As the ordinary temperature in an English room is 60°, and as at that temperature the weight of water per thousand cubic feet of saturated air is 0.82 lbs., as will be seen from Table V., the amount of water that should be contained in every thousand cubic feet of air supplied to a room is about 0.4 to 0.5 pounds.

Whatever the hygienic aspects of the matter may be, if the overcoming of this complaint as to dryness would in fact diminish the aversion to central heating apparatus, which is unfortunately so common, this alone would be sufficient justification for increased attention to this aspect of the work.

Moisture Emitted by Human Beings.

The quantity of moisture normally emitted by a human being is very variable. It depends on the individual. A person who drinks much fluid normally gives off more than the same individual would if he drank less, other conditions remaining constant. It is not necessarily true that of two individuals, he who drinks most gives off most vapour. Some persons have naturally drier skins than others. These persons dispose of a greater proportion of the surplus moisture through the excretory organs and less through their skins. The same individual gives off a much larger quantity of moisture when hard at work than when at rest, and much more on a hot day than on a cold one, and more in a dry atmosphere than in a damp one at the same temperature. In a warm and humid atmosphere much moisture is lost in active perspiration which soaks into the clothes and is again evaporated by the body heat.

The question is thus seen to be one of extreme complication and an exact treatment of the matter is an impossibility. It can only be dealt with by the method of averages. Such average quantities have been supplied by various authorities on hygiene. For an adult asleep the quantity is about .04 lb. per hour. For a man hard at work in a temperature of 75°, about 0.5 lb. The actual quantity emitted may vary any where between these two values.

For practical purposes one may take the average weight of vapour emitted per person to be 0.15 lb. at a temperature of 60° and at a mean humidity of 50%. This quantity is doubled when the temperature rises to 75%, with the relative humidity at the same figure.

It will be seen that the moisture normally emitted by one person will be sufficient to raise up to a humidity of 50% 730 cubic feet of air which has been warmed from 30° and 75% to 60°.

The Rate of Evaporation of Water under Various Conditions.

The evaporation of water exposed to air takes place at any temperature provided the surrounding air is not itself saturated. Even in the latter case the evaporation does actually take place, although it is counterbalanced by a corresponding amount of condensation (see page 45).

The rate of evaporation from any given surface depends on the amount of surface exposed to the air, on the temperature, and on the velocity of the air relative to the exposed surface. The amount of evaporation in any given case is directly proportional to the exposed surface, to the difference between the saturation pressure, corresponding to the temperature of the water, and the actual pressure of the water vapour in the air. Thus, the greater the temperature the greater is the rate of evaporation, other conditions remaining constant.

The rate thus varies inversely *with*, but not inversely *as* the pressure of the air present in the mixture. The quantity of evaporation is given by Box, according to the diagrams given on pages 51 and 53.

The humidity of air and the rate of evaporation have an important bearing on the practical problem of drying, as in laundries or for commercial purposes of various kinds.

Fuel Used in Drying.

In a well designed drying chamber one pound of fuel cannot be reckoned on as evaporating more than about 3 pounds of water at the outside. A more usual duty represents about half this amount. The exact figure depends a good deal on the efficiency of the arrangement and the way in which the air is passed over the surface, and the degree of humidity of the extraction air. About 45% of the total heat communicated to the air is generally utilised in the actual process of evaporation, the remainder being lost by conduction and in the actual heating of the air. The proportions given by Peclet are as follows :—

In evaporating moisture	45%
Carried off by the air	13%
Lost by radiation and conduction ...	42%
	100%

Means for Humidifying Air.

The simplest method of humidifying the air consists of one or more simple shallow cisterns or trays of water over the surface of which the air to be humidified passes. It will be seen from the calculations below that the surface of water needs to be considerable if any adequate amount of humidity is to be provided in a large volume of air.

In large installations on the Plenum system it is advisable to humidify the air to the requisite degree immediately after it has passed through the heating battery. The method commonly employed in this system consists of a number of flat trays of water, of which the free surface can be varied at will, the heat being supplied either by the heated air or by coils of pipes immersed in the water. If very hard water is used for this purpose it soon becomes covered with scale, and has to be constantly renewed and the vessels cleaned. It is very desirable that such vessels be enamelled to facilitate the cleansing process.

If horizontal surfaces are to be used they should be shallow tanks, one over the top of the other, having a sloping bottom, and means should be provided for lowering the level of water in each tank, either by means of adjustable ball floats or otherwise.

For smaller installations a more compact method of humidifying the air than by an open cistern of water, consists of a suitable length of hot pipe surface covered with canvas or other woven material, on which the requisite quantity of water is dripped. The arrangement may be made in many different ways.

Other methods are to introduce a stream of steam into the air. This method has, however, the disadvantage of producing a smell, similar to the soapy smell of a laundry wash-house, the cause of which it is not easy to ascertain. This method is often employed industrially.

Other means for humidifying the air are atomisers, which consist of means for directing a fine stream of water at high velocity on a flat surface in such a way that the water is scattered in every direction in a fine spray or mist, through which the heated air passes, taking up the moisture on the way. The disadvantage of this method is that it is not easily controlled, and it is easy completely to saturate the air or produce an excessive degree of humidity. A further disadvantage is that the small hole through which the stream of water impinges on the plate is apt to get stopped up. Such apparatus, therefore, needs continual attention.

Other means for securing the same object consist of a large wheel with horizontal axis rotating half immersed in a vessel of water. The air is moistened by passing between the flanges which are kept wet by the continuous rotation of the wheel.

Calculations in Humidity of Air.

The necessary calculations respecting the humidity of air for ventilation are required for the determination of—

(1) The amount of moisture to be evaporated per hour, taking count of the moisture emitted by human beings.
(2) The quantities of heat absorbed in the process.
(3) The size of evaporating surface.
(4) The quantity of relatively dry air necessary to prevent an undue rise in the degree of humidity in certain circumstances.

(1) The Amount of Evaporation Necessary.

The determination of the amount of moisture to be evaporated under given conditions is comparatively simple when the conditions are exactly known. Let it be required to determine the amount of moisture to be evaporated in order to bring 1,500,000 cubic feet per hour calculated at 35° and 75% humidity up to 70° having 50% humidity.

Here the absolute amount of moisture in 1,000 cubic feet of saturated air is found from the table to be 0.34 lbs. at 35° and 1.14 lbs. at 70°.

The air having relative humidity of 75% the absolute amount of moisture originally in 1,000 cubic feet is 0.75 × 0.34 lbs. = 0.255 lbs.

Hence the total amount in 1,500,000 cubic feet at the original temperature is

$$1{,}500 \times 0.75 \times 0.34 = 383 \text{ lbs.}$$

When this air is heated up to 70° it increases in volume to

$$1{,}500{,}000 \times \frac{461 + 70}{461 + 35} = 1{,}600{,}000$$

There are then 383 lbs. of moisture in 1,600,000 cubic feet of air, or 0.24 lbs. per 1,000 cubic feet.

But when saturated at this temperature it contains

$$1.14 \times 1{,}600 = 1{,}820 \text{ lbs.}$$

When at 50% of saturation the absolute quantity is 910 lbs. The amount to be evaporated is therefore

$$910 - 383 = 527 \text{ lbs.}$$

The diagram, Fig. 3*f*, shows clearly the behaviour of air as to relative humidity when heated, and a careful study of it is recommended. Problems such as the above can be directly solved by the proper use of this diagram.

Example 2.

If the quantity of air is required for the hourly ventilation of a hall containing 1,000 persons, determine whether any and if so how much moisture will be required to bring the air up to 50%.

This problem at once resolves itself into the question whether 1,000 persons will emit the required quantity of moisture in one hour. If they do this vapour may be assumed to be emitted uniformly. The quantity of moisture will be 1,000 × 0.15, or 150 lbs., leaving 377 lbs. to be supplied by the moistening apparatus.

Calculation of Quantities of Heat.

The quantity of vapour existing in any body of air has an influence on the specific heat of the air. The heat required to raise, say, 1,000 cubic feet of air at 60° F., with a pressure of 29.5 inches of mercury, and containing water vapour to the extent of, say, 60% of saturation, may be found by taking separately the quantity of heat necessary to raise the air, and that necessary to raise the water vapour to the desired temperature, as explained in Chapter IV. In making this calculation it must not be forgotten that by the process of heating the volume of the mixture increases, and in making any calculations in regard to heat supplied to air, it must always be clearly borne in mind what is the temperature at which the volume of air is to be calculated.

In the present example, for instance, in calculating the heat supply necessary to raise 1,000 cubic feet of moist air from 60° to 90°, careful note must be taken of whether 1,000 cubic feet is the volume of the mass of air at 60° or at 90°. It will be necessary in all such cases to calculate the weight both of the air and the water vapour.

It will be seen that the calculations of the heat necessary to raise the temperature of air are comparatively simple, when the total quantity of moisture in the air is not to be altered. Owing to the small absolute weight of the water vapour existing in a cubic foot of air, the heat required to raise the temperature of the water vapour as distinct from the air does not produce any very noteworthy alteration in the specific heat of dry air.

The case is quite otherwise, however, if, in addition to the heat required to raise the temperature of the mixture, heat must also be supplied to *evaporate* an additional quantity of water, so as to increase the total amount of water vapour in the air, with a view of increasing its humidity when raised to a higher temperature.

In making such calculations there are four separate quantities of heat to be considered:—

(1) That necessary to raise the quantity of dry air contained in the moisture (not the total weight of the mixture) through the given number of degrees.

(2) That necessary to raise the then existing quantity of vapour in the air through the same number of degrees.

(3) That necessary to raise the temperature of the water to be evaporated from its original temperature to the temperature desired.

(4) That necessary to evaporate the required weight of that water at that temperature for the purpose of further humidifying the air.

The relative magnitudes of these various quantities of heat can be seen from the various tables, and the following example illustrates the method of making the exact calculation, though approximate calculations are commonly near enough for the purpose.

Required the quantity of heat necessary to raise 1,000 cubic feet at 32° F. and 75% saturation up to 90° F. and 50% saturation, the pressure being 27 inches of mercury.

From Table V., column 7, we see that the pressure of water vapour is at 75% saturation at 32° F., 0.181 inches of mercury. Deducting this from the total atmospheric pressure we get the actual pressure of air equal 26.819

inches of mercury; the weight of this air will be seen from Table V., column 3, to be

$$80.7 \times \frac{26.819}{29.92} = 72.4 \text{ lbs.}$$

In order to raise this air to a temperature of 90° F., at which temperature its volume at the same pressure will be $1{,}000 \times \frac{551}{493} = 1{,}120$ cubic feet, the amount of heat required will be $72.4 \times 0.238 \times 58 = 999$ B.T.U. The total weight of water vapour in the air at this time will be $1.12 \times 2.12 \times 0.5 = 1.18$ pounds.

In order to raise the temperature of this vapour up to 90° F. the amount of heat required will be $1.18 \times 0.475 \times 58 = 32.5$ B.T.U.

Now at 90° F. and 50% saturation, which is its final condition, the total quantity of water vapour necessary to humidify 1,120 cubic feet of space will, as shown above, be 1.18 pounds. The amount of water vapour actually existing in the air before its humidification will be $0.304 \times 0.75 = .228$ pounds. Therefore the amount of water required to be evaporated is $(1.18 - 0.228) = .952$ pounds. The amount of heat necessary to evaporate that water is $.952 \times 1{,}084 = 1{,}035$ B.T.U.

Careful note must be taken of the fact that it is the *space* which requires to be saturated and not the *weight of air.*

From this example will be seen the relative importance of the quantity of heat necessary for the purpose of evaporation as distinct from the mere raising of the temperature.

The reverse calculations are equally important and perhaps even more complicated, for here we have to take account of the heat given up by the vapour in condensing to water.

For instance, suppose that a thousand cubic feet measured at 90° F. and having a humidity of 80% has to be reduced in temperature to 80° F. and a humidity of 60% is required, what amount of heat must be taken from the air in order to produce this result?

Here the heat given off by a thousand cubic feet of air weighing about 71.3 pounds, reduced from 90° F. to 80° F., $= 71.3 \times 0.238 \times 10 = 170$ B.T.U.

The amount of vapour contained in a thousand cubic feet of space at 90° F. and 80% saturation is $2.12 \times 0.8 = 1.7$.

In order to reduce this vapour to a temperature of 80° F. only $1.7 \times 10 \times 0.475 = 8.1$ B.T.U. must be taken from it. The amount of water vapour in 980 cubic feet of the mixture at 80° F. and 60% saturation will be $0.98 \times 1.56 \times 0.6 = 0.92$ pounds.

It is therefore evident that $1.7 - 0.92 = 0.78$ pounds of water vapour must have been condensed during the cooling process, which will give up

$0.78 \times 1{,}055 = 820$ B.T.U. The reduction in temperature of the water vapour will only be responsible for 8.1 B.T.U.

The total result of the calculation is therefore that

$$170 + 8.1 + 820 = 998$$

B.T.U. must be abstracted from the air in order to reduce its temperature to the desired condition.

Calculations are sometimes required in order to ascertain the amount of air which must be introduced at a given temperature and having a given degree of saturation in order to prevent the relative humidity of the air in a room occupied by a certain number of human beings from rising above a certain maximum.

For instance, assume that the outer air is at 60° F. having a humidity of 50%. A hall of 120,000 cubic feet is occupied by 400 persons, it is required to find how much of the outer air must be propelled into the hall in order to prevent the humidity therein from rising at a temperature of 70° to a point not exceeding 65%.

In this case the quantity of moisture in a thousand cubic feet of air at 70° F. and having a humidity of 65% would be found to be

$1.14 \times 0.65 = 0.74$ pounds per thousand cubic feet.

Let x be the quantity of air in cubic feet per hour to be propelled into the hall calculated at 70°. Then x is also the volume of air expelled through the outlets, the expelled air being in the given condition as to moisture, must carry with it 0.74 pounds of moisture per thousand cubic feet.

The introduction of x cubic feet of air calculated at 70° implies that

$$x \times \frac{461 + 60}{461 + 70} = .980\,x$$

cubic feet of air at the *outside* temperature, 60°, have been propelled into the hall, this having a humidity of 50%, contains $0.82 \times 0.5 = 0.41$ pounds per thousand cubic feet in its composition, and therefore the introduction of this quantity of air x into the hall involves bringing into the hall

$$0.98\,x \times .00041$$

pounds of moisture.

The persons in the hall will emit, say, 0.15 pounds of moisture per head per hour, the total weight of moisture emitted being therefore

$400 \times 0.15 = 60$ pounds per hour.

Hence we have the equation

$$x \times 0.00074 = 0.98 \times 0.00041\,x + 60$$

from which the value of x is found to be

$$\frac{60}{0.000740 - .000402} = 177{,}000$$

In practice the quantity of moisture in the outside air cannot be exactly determined because of the diurnal variations in the relative humidity, so that in all practical calculations made for this purpose a high average outside humidity must be assumed, say, 80% of saturation.

Appropriate values can easily be determined from the Meteorological Office diagrams given on page 57. The moisture given off by the burning of gas or stoves, or other means adopted to light and perhaps to warm the room may have to be taken into the calculation in the same way. These depend on the special circumstances and a little judicious care is all that is necessary to make the calculation correct.

CHAPTER XXII.

METHODS OF PRODUCING CIRCULATION IN SYSTEMS OF WATER PIPES.

THE principles governing the flow of water in pipes when propelled by definite heads have already been explained. It is now necessary to explain the practical methods whereby a difference of head is maintained in a heating installation, resulting in the establishment of a constant circulation through systems of radiators.

Speaking generally, all these systems have the same object, namely, to maintain in the flow or hot pipe a greater circulating pressure (see page 82) than in the return or cold pipe, so that when a connection is made from the flow pipe to a radiator, and from the radiator to the return pipe, a flow of hot water is established and maintained through the radiator or series of radiators.

The method of application of the main principles is dictated by the necessity for securing uniformity as far as possible in the flow through radiators fixed in different positions.

Just as in the case of air circulation, the methods adopted are what may be called, (1) the natural system, and (2) more or less artificial systems of producing this difference of pressure. Of these, the natural system which is adopted in the ordinary low pressure gravity method is by far the most generally used, and also by far the most difficult to understand and to proportion correctly.

We shall therefore explain this principle in some detail.

If a pipe is completely filled with water it is obvious that that water will remain at rest, or if not initially at rest will rapidly come to rest in the pipe unless there is a continual supply of power pressing the water along the pipe. The movement of water along a pipe necessarily involves a continual destruction of mechanical power, for when water is maintained in motion along a pipe, every inch of the pipe exerts a certain frictional force on the water tending to stop the flow. The supply of power to the water tending to maintain the flow must be such as exactly to counterbalance the frictional force tending to stop it. Only in these conditions will a steady uniform flow of water be maintained.

If the force tending to move the water is at any moment or at any point greater than the total frictional force resisting its motion, the velocity of the water in the pipe will be continually increased until frictional forces are called into play, of magnitude exactly equal to the propelling force.

Since the pipe is supposed full of water the condition controlling all changes of velocity within any part of the circuit is that the rate of increase of volumetric velocity throughout the entire system shall be constant. The water cannot increase in velocity in one part of the system without a simultaneous increase in all parts. Expressed in another way, any increase in circulating power generated in one part of the system is at once transmitted and distributed throughout the entire system.

Once the momentum has been produced in the mass of water, no more energy has to be expended merely in maintaining the momentum, for apart from the retarding force, due to friction, the momentum would be maintained of itself, according to Newton's law of motion.

In any case the actual absorption of power necessary to maintain a low velocity in a pipe, such as heating engineers are accustomed to employ, is very minute. Such as it is, the effect of maintaining that power is not lost, as the power is itself converted by friction into heat. This is of course utilised along with the remainder of the heat existing in the water. The question, therefore, of the actual absorption of power, trifling in itself, is of no moment whatever, except in so far as it costs more to produce one thermal unit by friction or its equivalent from any source of mechanical power actuated by any kind of a prime mover, than it does to produce one thermal unit by the ordinary means chiefly utilised by the heating engineer.

However, given the necessary power for driving the water through the pipes, whatever the source of this power may be, what the heating engineer is chiefly concerned with is to see that the pipes are of such dimensions and arranged in such a manner that the available power can force through the pipes sufficient water to supply the amount of heat required at the various points to which it is led.

The Principles of Circulation on the Natural System.

Consider a small installation similar to that shown on Fig. 112 consisting of an expansion tank, some kind of heater *B*, and a system of large pipes. Suppose this apparatus is filled with water up to the level *AA*, the water being all cold. It is clear from the laws of hydrostatics that the water will rise to the same level in the expansion tank and in the two pipes *BF* and *CR* at *D* and *E*. There is then a complete balance of pressures in every part of the apparatus, and the absolute pressure at any point can be determined, knowing the common temperature and the vertical distance of the point below the common surface or water line.

Pressure in pounds weight per square foot at any point = depth in feet × density in pounds weight per cubic foot = $h \times D$. A diagram showing the values of these pressures is shown in Fig. 112, plotted in each case on the axis of the pipe as base. For clearness sake these pressures are in this and the subsequent diagrams shown roughly to scale, in feet of water gauge. The water in the gauge is assumed in all cases at the standard temperature, 62° F., which is also assumed to be the temperature of the water in the expansion tank.

If now the water in the pipes F and R be heated to different degrees, the valve V_1 being shut, the water in them will expand and become of

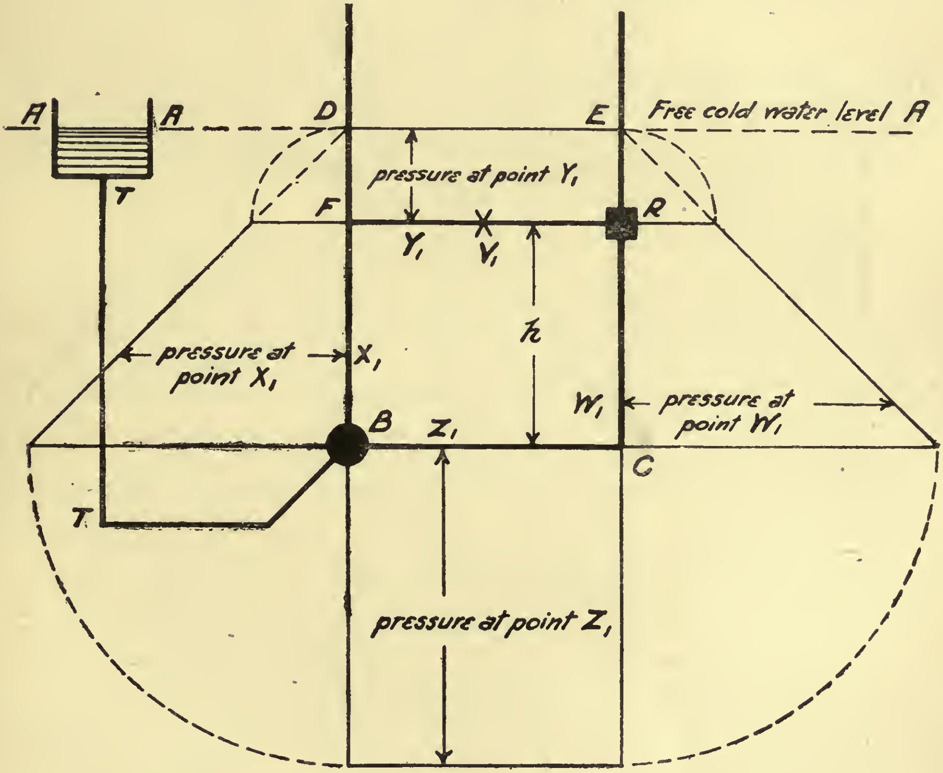


DIAGRAM SHEWING THE STATIC PRESSURES IN A SYSTEM OF PIPES WITH WATER ALL COLD.
FIG. 112.

less density than the water in the pipe *T*. This state of things is shown in Fig. 113. In order, therefore, that the water columns BA_f and CA_r shall produce the same pressure as before at the bottom of those pipes, and so be in equilibrium with the column *T*, it is clearly necessary that the water must stand at a higher level in pipes BA_f and CA_r than the level *AA*, although the pressure at the bottom of the pipes is the same as before. Let these new "free water levels" be indicated by points *D* and *E*, as shown in Fig. 113.

Of course, during the heating no water either enters or leaves either pipe *F* or *R*. The water already in these pipes is merely expanded so

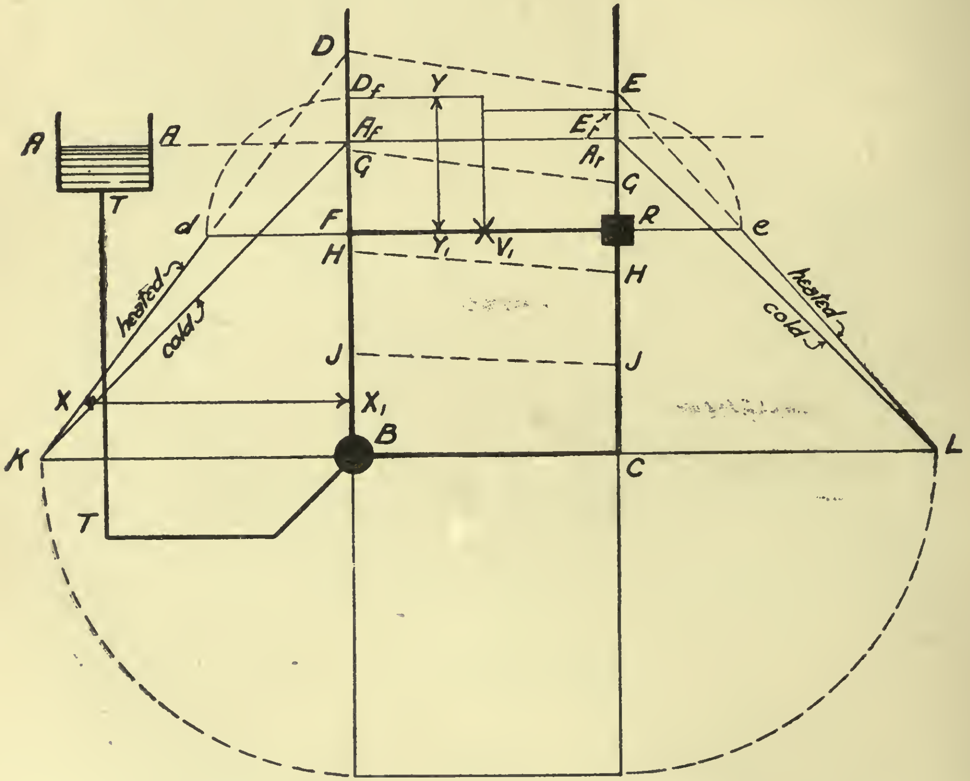


DIAGRAM SHOWING A COMPARISON OF PRESSURES
IN A SYSTEM OF PIPES (1) WHEN WATER IS COLD.
(2) WHEN RISERS ARE DIFFERENTIALLY HEATED.
A. UPPER VALVE CLOSED.
FIG. 113.

that the pressure which is produced by the dead weight of the water is not altered by the expansion.

Assume, for purposes of illustration, that the pipe *BD* is heated to 170°, and pipe *CE* to 130°, the temperature in *T*, as before, being 62°, and assume that the height from the water level *AA* to the level *BC* is 30 feet. Then calculated from the table it is evident that point *D* will stand 9.42″ above the level *AA*, and point *E* 4.74″ above that level.

There will then again be a perfect equilibrium in the system, so that there is no water flowing in any direction so long as these temperatures are maintained constant.

Now compare the state of pressure in the pipes *BF* and *RC*.

It is evident that point *D* is at the same pressure as the point *E*, since they are both open to the atmosphere. It is also evident that since no water is flowing, the pressure at point *C* is the same as the pressure at point *B*.

Now draw lines *GG*, *HH*, etc., between corresponding pairs of points, dividing *BD* and *CE* in equal proportions. Each pair of points are obviously points of equal pressure in the two columns. The dotted lines merely indicate the identity of pressure at their two ends.

DE will be a sloping line, sloping downwards from *D* to *E*. *BC* will be a horizontal line. The lines of equal pressure will slope more and more the higher they are, being very nearly horizontal at a low level, and very nearly parallel to *DE* when they are near that level.

In order further to illustrate this, the actual pressures when the water is all cold are shown at all parts of the system, plotted on the axis of the pipe itself as base. Also the pressures are shown after the water becomes heated, the respective lines being labelled. Note that for uniformity in this and all succeeding diagrams pressures are indicated as heights of water column *at a temperature* 62°, which would produce the actual pressures found at that point.

Thus, the lengths of the lines $X_1 X$, $Y_1 Y$ represent to scale the height of water columns at a temperature of 62°, which would produce the pressures actually found at X_1 and Y_1 when the water in *BD* is heated to 170°, and similarly for the return side.

It is to be carefully noted, in order thoroughly to understand these very important diagrams, that the reason why the points D_t and E_r do not coincide respectively with *D* and *E* is that the line FD_t represents the pressure at *F* in length of water column *at a temperature of* 62°, whereas the line *FD* represents the same pressure in length of water column at a temperature of 170° F. The lines *KD*, *LE* are not inclined exactly at 45°, though KA_t, LA_r are. All these diagrams should be closely studied.

Note also that the greater the density of water, that is, the lower its temperature in any riser, the more will the corresponding line of pressure slope away from the vertical. The line representing the pressures in a hot riser will be more nearly vertical than that representing the pressures in a cold riser.

Now suppose that the valve V_1 in the horizontal pipe *FR*, connected between *F* and *R* at any level, is opened. Let the end *R* be provided with a cooling coil or radiator of such power as to maintain the temperature of the falling pipe *RC* exactly at the required temperature, the boiler *B* performing a similar service for the riser *BF*.

The water must flow through the cross pipe from *F* to *R*, *because the pressure Fd, at the F end, is greater than that Re at the R* end, and the amount of water flowing through will depend on the excess of pressure on the *F* side. There will thus be less water flowing through if the connection is made lower down and more water if higher up, *because the difference of pressure at the same level is greater higher up than it is lower down*, as shown by the sloping dotted lines.

When the upper valve is closed, therefore, the state of pressure in the entire system is shown by the system of lines on Fig. 113.

Now suppose that a closed valve V_2 is fixed in pipe *BC*, the upper valve V_1 being left open. The state of pressures would then be as shown on Fig. 114.

In this case it is clear that the upper connection would serve to equalise approximately the water levels in the two columns, and to equalise them exactly if the densities in the upper parts *FD*, RE_d of the two columns are the same. But if these densities are different, there would be a slight difference of level between *D* and *E* , due only to the relative expansions of the short columns *FD* and *RE* . For purposes of the present explanation this slight difference of level will be neglected. The difference of pressure which in the former case existed between *F* and *R* would then be transferred to the bottom of the pipes and would be shown by a difference in pressure on the two sides of the valve V_2, as shown on the diagram. Points of equal pressure in the pipes *BF*, *CR* would then be as shown at *GG*, *HH*, *JJ*, etc.

Now let Fig. 115 represent the case where both valves are left open. In consequence of column *CR* being always heavier than *BF* there will then be a continual circulation of water, say, *Q* pounds per hour, maintained through the system of pipes on the same principle as that on which an overshot water wheel goes round when the buckets on one side are kept full and on the other empty.

Or to vary the simile, the hot water in the column *BF* is forced upwards for exactly the same reason that a vertical cylinder of cork is forced upwards when it is immersed in water, namely, that the upward vertical force due to the hydraulic pressure on the bottom circular end of the cork exceeds the downward force exerted on the top circular end by a difference which is greater than the weight of the cork, while the horizontal forces on the cylindrical bounding surfaces of the cork are in equilibrium (see page 11).

Just so here conceive the cylindrical mass of water in *BF* as a mass of substance lighter than the liquid in pipe *BCR*. There are only three vertical forces acting on this column *BF*, (1) the pull of gravity on it, *i.e.*, its weight; (2) the downward force on the end *F*, due to the pressure at that section; (3) the upward force on its end *B*.

Then just as in the case of the cork, the total upward force due to the hydraulic pressure arising from the column *CE* of dense liquid acting on the circular end *B* of the lighter liquid exceeds the downward force on the end *F* of the same column, by a difference which is greater than the weight of the column *BF*. The resultant of the forces on the column *BF* is therefore upwards.

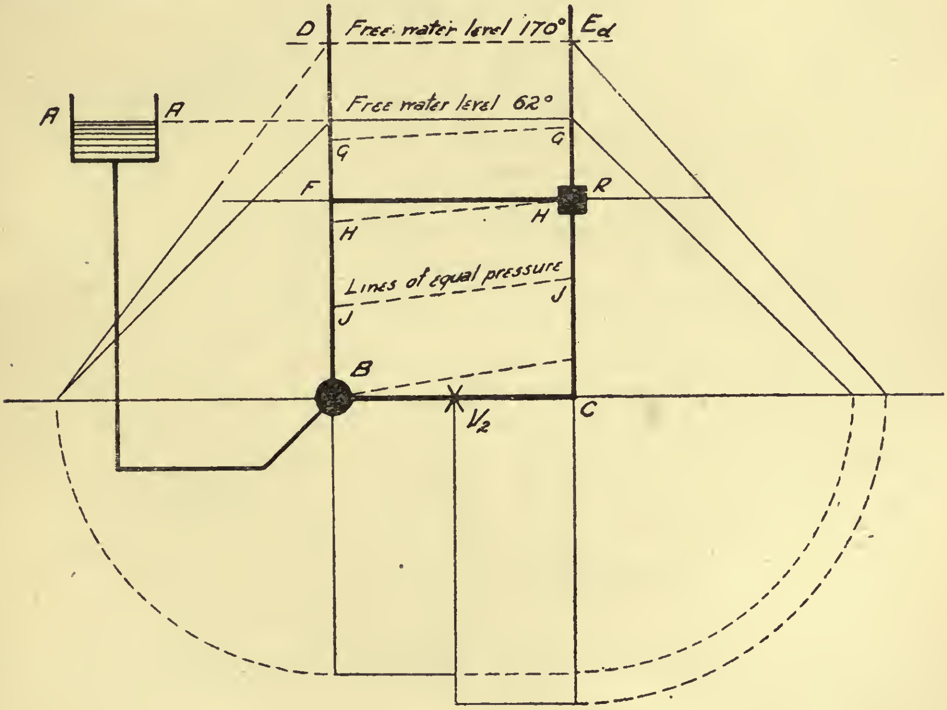


DIAGRAM SHOWING A COMPARISON OF PRESSURES
IN A SYSTEM OF PIPES (1) WHEN WATER IS COLD.
(2) WHEN RISERS DIFFERENTIALLY HEATED.
B. LOWER VALVE CLOSED.

FIG. 114.

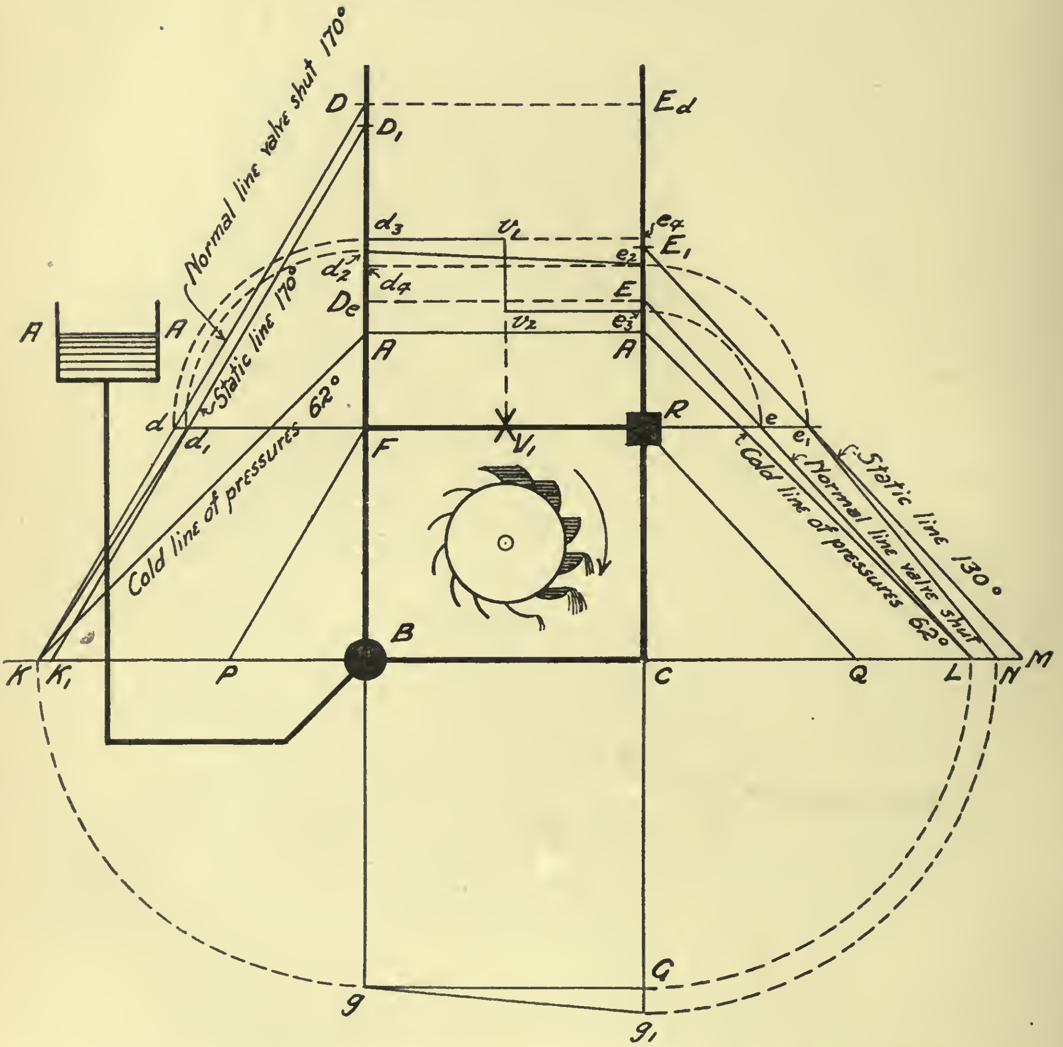


DIAGRAM SHOWING COMPARISON OF PRESSURES IN A SYSTEM OF PIPES

(1) WHEN WATER IS COLD.
(2) RISERS DIFFERENTIALLY HEATED, UPPER VALVE CLOSED.
(3) RISERS DIFFERENTIALLY HEATED AND CIRCULATION PROCEEDING.

FIG. 115.

The consequence of the flow of water will be an entire rearrangement of the pressures throughout the system. This rearrangement is seen in the diagram, Fig. 115, which should be carefully studied. In this diagram AA is the water level of the expansion tank when the water is all cold, the diagram of pressures throughout the system is $KAAL$, as before. When the risers are differentially heated so that D represents, as before, the free water level in the hot riser, and E the free water level in the return, then when valve V_1 is shut the diagram of pressures is $K\ d\ d_3\ v_1\ v_2\ e_3\ e\ L\ G\ g$.

When the valve V_1 is opened the rearrangement of pressures takes place as follows :—

Here the free water level D falls to D_1, and the free water level E will rise to E_1. The flow of water along the upper pipe FR will keep the levels D_1 and E_1 more nearly equal than they would be if the valve V_1 were closed, but not so nearly equal as if valve V_2 in Fig. 114 were closed.

The difference which will be maintained in fact is such as is necessary to force the quantity of water Q pounds per hour through the pipe FR. The pressures and water levels will adjust themselves in such a way that although level D will fall to D_1, the absolute pressure at B, where the feed pipe joins the system, is always constant, and a difference of circulating pressure (see page 82), whose value is $d_1\ d = d_2\ d_3$, will be generated between points B and F, sufficient to force the quantity of water Q pounds per hour up through the rising pipe BF.

The level D_1 will be sufficiently high above E_1, that the same quantity of water Q will be forced through the pipe FR by the pressure $d_4\ d_2$. The level E_1 will be sufficiently high above the normal E that the same quantity of water Q will be forced per hour down the pipe RC by the pressure NM, and the pressure Cg_1 at C will be higher than it would be in a static condition by such a pressure Gg_1 as is necessary to force a quantity Q pounds per hour along the pipe CB back to the boiler, owing, of course, to the raising of the free water level E to E_1 by the access of pressure transmitted by the pipe FR.

The actual pressures throughout the system when the water is flowing expressed in terms of water column at 62° is then

$$K\ d_1\ d_2\ e_2\ e_1\ N\ g_1\ g$$

Now if the free water level in CE stood at E_1 when the water was at rest, and if the temperature in CE were the same as before the valve was opened, the line of pressures in CE would be $E_1\ M$, where $E_1\ M$ is parallel to EL. This line we shall call the "static line of pressures with a free water level E_1."

Similarly, if the free water level in the rising pipe stood at D_1 when the water was at rest, the static line of pressures would be $D_1\ K_1$ where $D_1\ K_1$ is parallel to DK.

The adjustment of pressures, in fact, represents Nature's automatic solution of a long series of simultaneous equations. The only part of the system at which the pressure will remain unaltered will be at the boiler or other part to which the expansion tank is connected. The remaining pressures will automatically adjust themselves in the manner just described, which is illustrated on the diagram, Fig. 115.

It is therefore clear that the total difference in pressure which is producing circulation in the entire system is $K K_1 + d_4 d_2 + N M + G g_1 = d_4 d_3 + e_3 e_2 = e_3 e_4 = Fd_3 - Re_3 = Fd - Re = (B K - B P) - (C L - C Q) = C Q - B P$ (since $C L = B K$) = the difference in pressure due to the difference in density of columns *CR*, *BF*.

This total available head is gradually used up right throughout the system. When the water is flowing, there are no two points in the entire system between which there exists so great a difference of circulating pressure as $e_3 e_4$, yet this total head is actually producing the circulation.

A consideration of this diagram will explain many peculiarities about a hot water circulation.

Consider for instance the case of a short circuit. Suppose there are several pipes one above the other connected between pipes *BF* and *CR*. It is well known in practice that if the upper pipe is too large, the lower pipes will be "short circuited." The reason for this will be clearly understood from consideration of the diagram.

The larger the pipe *FR* is the more nearly will the level D_1 approximate to level E_1, for the smaller will be the pressure necessary to force the quantity of water *Q* through the pipe. Therefore the higher the pressure will be all the way down the pipe *RC*, and the lower it will be all the way up the pipe *BF*. As *RC* is cooler than *BF* the line $E_1 M$ stands further away from the vertical than does $D_1 K_1$.

If then the second cross pipe is fixed (as it may easily be) at a level at which the actual pressure in pipe *CR* is greater than that in pipe *BF*, then the water will actually flow *from* the pipe *RC* to the pipe BF. If the height is so chosen that the two pressures at the same level are the same, the stagnation in the pipe will be complete, and no water whatever will flow through the junction.

If the level is so chosen that the pressure on the *F* side is greater than on the *R* side, the water will circulate according to the difference.

Similarly, the pipe being fixed at any particular level it will be possible, by altering the amount of opening of the valve V_1 in pipe *FR*, so to alter the distance between the free levels $D_1 E_1$, as to alter the direction in which the water flows in the lower pipe, or to stop it altogether.

Thus, the question whether there would be any or what circulation in any radiator in a system of pipes depends simply on the pressures at the beginning and end of those two pipes, *the pressure being measured when the water is flowing*. This aspect of circulation will be further considered in a later chapter.

If there is no circulation to some particular radiator, the practical cause of it may be obscure, but of the physical cause there is no possible doubt, namely, that owing to some such reason as above described, the pressures at the beginning and end of the pipe are the same. Any cause, therefore, which will disturb that balance will be sufficient to start circulation in the pipe.

Other Methods of Producing Circulation.

There are other more artificial methods of causing hot water to flow through pipes which are all incomparably easier to understand than the natural system explained above.

All these methods consist of producing and maintaining a head of water greater on the flow pipe than on the return pipe. Any simple method which will bring about this end may be employed for maintaining the excess of pressure which is a fundamental necessity for the continued circulation.

Common methods for causing water to flow in pipes can be and are extensively used, such as reciprocating pumps, centrifugal pumps, injectors. There are also several other well-known means or systems which have been specially devised for provoking a circulation, such as the "Cable" or "Barker" system, the Æro system, the Reck system, the Bruckner system, the Beck system, and others. All these systems produce a difference in pressure between the flow and the return pipe fixed at the same level, much greater than the difference produced by the ordinary difference of specific gravity of the water, which is the fundamental principle on which the ordinary or natural system of circulation works.

Analysing these different methods, it will be found that the result of each one is merely to increase the pressure in the flow pipe, or to decrease that in the return pipe, or both.

1. *Reciprocating Pump.*—Functionally this is in reality a means of alternately expanding and contracting chambers in connection with the flow and return pipe respectively, combined with non-return valves, such that the water can only flow one way round. The expansion decreases the pressure in the return pipe by taking water from it, and the contraction increases the pressure in the flow pipe by forcing more water into it.

2. *Centrifugal Pump.*—This also produces a suction on the return pipe, and an increased pressure on the flow pipe, by means of the rotation of a disc or fan within the water itself, in such a way as to force water from the return pipe to the flow pipe by centrifugal action.

3. *An Injector* produces a similar result by an induced current formed by blowing a jet of steam into the water at a high velocity.

4. *On the "Cable" or Barker system* the pressure in the return pipe is reduced by connecting the return pipe with a chamber containing steam at a very low pressure, which condenses the steam and heats the water at the same time. By producing and maintaining a partial vacuum in this chamber the pressure in the return pipe is reduced. The apparatus then lifts the heated water by steam pressure to a tank at a higher level, which tank is in connection with the flow pipe, thereby increasing the pressure in the flow pipe, and simultaneously decreasing that in the return pipe.

The Reck system decreases the pressure in the return pipe only, by blowing steam into the rising portion of the flow pipe, thereby substituting for a solid column of water an emulsion or mixture of steam and water, thereby raising point *D*.

The Bruckner system effects the same end in a somewhat different way, by boiling the water in the rising portion of the main flow pipe, thereby generating steam bubbles in it and making it lighter.

The effect of any of these methods in producing rapid circulation or accelerating the circulation, depends on the degree to which the pressure in the flow pipe is increased over that in the return pipe, and the dimensions of the pipes are determined to correspond with the excess of pressure.

CHAPTER XXIII.

ON THE DETERMINATION OF PIPE SIZES FOR A HOT WATER INSTALLATION.

I.—THEORETICAL PRINCIPLE.

THE broad principle on which a low pressure hot water apparatus works has been explained in detail in the last chapter.

The emission of heat from radiators and the quantity of water necessary to supply this heat have also been fully treated, as well as the laws giving the amount of fluid friction in pipes of various sizes when definite quantities of water per hour pass through them.

It now remains to explain the method whereby all these principles may be combined into a determination of the necessary sizes of pipe to supply certain quantities of heat at different points of a hot water installation.

Definitions.

The terms used in this discussion will first be explained.

The arrangement of the circulating pipes may be made according to one or more different systems. Any or all of these systems may exist in the same apparatus.

The usual systems of circulation may be classified into four main sub-divisions—

(1) The "two-pipe" system, (*a*) overhead, (*b*) underground.
(2) The "single-pipe" system, (*a*) overhead, (*b*) underground.
(3) The "one-pipe-series" system, and
(4) The "circuit-pipe" system.

But this classification is not, and no classification can be, an exclusive or perfect one. There are a great number of different methods of arranging pipes; indeed, strictly speaking, there are an infinite number of different systems, any of which, if properly designed, will maintain a satisfactory circulation. Many of them combine the features of some or all of the above main sub-divisions.

(1) All systems necessarily comprise a main flow pipe and a main return pipe, and in so far all are "two-pipe" systems.

The main characteristic of what we shall call the "two-pipe system" is that the same water only passes through one radiator, after which it returns to the boiler.

(2) On what we shall call the "single-pipe system," however, any particle of water may pass through several radiators, one after the other. In most "single-pipe systems" the temperature of the water, after passing through the first radiator, is reinforced by mixing it with hotter water which has not passed through any radiator.

(3) In the "one-pipe-series system," however, radiators are connected on the one-pipe system in series, in such a way that the whole of the water after passing through the first radiator passes through the second, and after that perhaps through a third, and so on. In this case the two radiators and the pipe between them may be regarded as in reality different parts of one radiator, which may be assumed to have been cut in two or more parts, and the calculations become similar to the two-pipe system, except that the resistance of the radiator is increased by the addition of the connecting pipes.

(4) The "circuit system" is in reality several one-pipe-series systems combined in one installation.

Water generally leaves the boiler in one "principal main," being sub-divided in its passage into "branch mains," these again branching into "circuits" which lead to the various radiators.

A "flow pipe" is one which carries hot water to a radiator or coil. A "return pipe" is one which carries water from a radiator to the boiler. On one-pipe systems the same pipe is frequently both a flow and a return, being a flow pipe with respect to one radiator and a return pipe with respect to another.

A "circulation" consists of any complete and continuous series of pipes through which water flows in the same direction from the flow outlet from the boiler to the return inlet to the boiler, passing through one or more radiators or coils on the way. Each circulation will, in single-pipe systems, feed more than one radiator, and the sum of all circulations in an installation must comprise every pipe and radiator in the entire installation.

On the pure two-pipe system there is a separate "circulation" for each radiator, though parts of each circulation are generally, of course, common to more than one circulation.

A "*circuit pipe*" will be spoken of as the connection from a flow main to a return main, passing in its course through one or more radiators in series, but *being unbranched*, except in cases where *both* flow and return connections are made from the pipe to one or more radiators, as in the one-pipe system.

A "*main pipe*" will be spoken of as one which feeds two or more circuits. A main pipe may be either a "principal main" or a "branch main." If the water flowing through any pipe is divided into two or more definite streams which do not meet again until they enter the return, that pipe is called a main pipe, otherwise it is a circuit pipe.

A "*principal main*" is one main connected to the boiler which feeds two or more "branch mains."

A "*branch main*" may be either a "primary branch," a "secondary branch," or a "tertiary branch," etc.

The "travel" of any circulation, radiator, or pipe, is the *greatest* total distance which *any of the* water flowing through that pipe or radiator has to go between leaving the boiler at the flow outlet and arriving at the return inlet.

General Principles.

Considering a building as a whole, the object aimed at is, of course, to design a system of pipes which will maintain such a circulation of water from the boiler as will deliver the requisite quantity of heat at the various points.

If a quantity of water Q pounds passes through the boiler per hour, and in its passage is raised in temperature from t_r to t_f, the temperature falling again from t_f to t_r during its passage round the system, then the required quantity of heat H B.T.U. will be delivered into the building, only if the condition is fulfilled that

$$H = Q\ (t_f - t_r)$$

The attainment of the object, therefore, depends on securing two results.

(1) That the quantity Q pounds per hour shall pass through the boiler.

(2) That in passing round the system its temperature shall fall from t_f to t_r, or in other words, that in passing through the boiler it shall be raised from t_r to t_f, and if these two conditions are secured, the *total* delivery of heat into the building is equal to the required H B.T.U. per hour.

Single Circuit Installations.

In the case of a simple single circuit, where all the pipes and radiators are in one room, the exact quantity of water does not matter much, nor the exact value of its temperature, provided always that

$$Q\ (t_f - t_r)$$

is numerically equal to the quantity of heat required.

The velocity actually attained in any pipe, the temperature of the water, and the diameter of the pipe, are interdependent, according to a most complicated relation.

Other things being equal, the greater the velocity of the water the less will be its drop in temperature between any two points of the circuit. But the circulating power forcing the water through any complete circulation depends directly on the difference in temperature between the hot rising column and the relatively cool falling column.

Combining the above two statements, it will be seen that the greater the velocity of the water the less the circulating force will be.

The larger the pipe the more heat will be lost from it at a given temperature, and the greater the velocity with which the water will flow through the pipe under the influence of a given circulating head. But, as above shown, the greater the velocity of the water the smaller is the difference of temperature, and the less the circulating power. The relation between all these variables is thus seen to be excessively complicated.

Under the operation of natural forces the velocity of the water will increase up to such a point as to produce a difference in temperature between flow and return, which will be just sufficient to maintain the circulating head necessary to overcome the friction in a pipe of the given size when the water is moving at that particular velocity. Any correct system of designing the pipes must provide for balancing all these variable quantities. It is essentially a problem of considerable complexity.

All the above statements are true, whatever the size of the pipe, and in commencing to design such a simple system as this we are given only the quantity of heat required. The sizes of the pipe are not known to begin with, and we can therefore have no idea of the amount of heat that would be lost on the way to the radiator.

It will be evident, therefore, that even for a single circuit the equations which would determine directly the velocity of the water corresponding to any given size of pipe, or the size of the pipe for any given quantity of heat, would be of considerable complexity. It would be altogether out of the question to require the solution of such equations as a preliminary to designing a system.

Compound Circuit.

The case of a compound circuit is more complicated still. In this case the object aimed at is not only to secure that the *total* quantity Q pounds per hour shall pass through the boiler with a suitable difference in temperature between flow and return, but also that the total quantity of heat so delivered into the building shall be properly distributed in the correct proportions to the required points. In other words, that if H_1

B.T.U. per hour are required *in any room*, then a quantity Q_1 lbs. of water per hour must flow through that room, and its temperature shall fall from t_{f1} to t_{r1}, the relation between the values being such that

$$H_1 = Q_1 \ (t_{f1} - t_{r1})$$

The problem is to secure this result for *each room* supplied by the installation. If it is not secured for any room, then that room cannot receive its designed quantity of heat.

The exact temperature at which the water enters and leaves the room would not of itself be of great importance *so far as that particular room was concerned*, provided the quantity of water flowing through it per hour is great enough to leave behind the required quantity of heat in cooling between the two temperatures.

But if one return branch pours water into the return main at a much higher temperature than the others, this hot water will flow from the main return into the return branch leading from some other radiator, and as the circulating power of that radiator depends on the difference of temperature between its own flow and return, this hot water will check, or in extreme cases, may entirely stop the weaker circulations.

It is therefore very desirable that the water in the return pipe from each circulation may be at approximately the same temperature. This is the governing condition which determines the uniformity of the result, and its attainment is the main object to be aimed at in designing a compound system of pipes.

In order to secure it, it is obviously necessary that the quantity of water flowing through each circulation should be approximately correct. If too much flows through any particular circuit, the return temperature from that circuit will be too high, and if too little, the return temperature will be too low.

In each circulation of a compound system the force propelling the water through that particular circulation is produced by the greater weight of its falling return column over that of the rising column, and it is true of each circulation that the velocities of the water must adjust themselves so that the total motive force in that circulation will be exactly balanced by the friction that is called into play, or in other words that the velocity will continually increase until this exact balance does exist.

This is a fundamental law of nature, and is necessarily and exactly true for each circulation, whether the pipes are properly proportioned or not.

There is nothing theoretical about the above proposition, it is an absolutely certain deduction from the fundamental law of nature that action and reaction are equal and opposite.

Whatever is uncertain about the flow of water in a hot water apparatus, this much is absolutely certain, that in all cases the force generating the velocity is equal to the resistance produced by that motion. Its truth does not depend on the accuracy of the particular formula adopted for the resistance. If any formula does not agree with deductions made from accurate observations made on this point, then that formula is incorrect.

It will be clear from the preceding that the object aimed at in any scientific system of pipe sizing is to proportion the size of pipes to the available circulating pressure, and the required quantity of water. The chief difficulty in understanding how this can be done is that in the great majority of circulations every main pipe must be designed to feed radiators at different levels, and therefore through circuits having different circulating powers all pulling on the same main.

To be satisfactory the system must be such that it will supply an adequate quantity of water to every radiator to which it is connected. When, as frequently happens, the same pipe feeds radiators at a high level as well as at a low level, the question arises what will be the effect on a flow pipe divided between two or more branches, of the existence in one of them of a high circulating power and in the other of a low circulating power.

It is clear that if the sizes of the main pipes feeding the radiator with a high circulating power are calculated according to the position of this latter radiator, they will be smaller than if they are calculated with reference to the position of the lower radiator. But as the main object to be aimed at is to get a sufficient quantity of water through the pipes, the question arises, does the increased circulating power of the higher radiator help or hinder the flow of water to the lower radiator?

On the one hand it certainly helps in pulling an increased quantity of water through the common mains, and at first sight it would seem to be an advantage rather than otherwise to another radiator also fed from those mains, in that it increases the power of the circulation of water through part of the mains which feed it.

But a further analysis of the matter on the method adopted in Chapter XX. will show that the presence of the highly placed radiator on the same mains as the low placed one is not only of no assistance to the lowly placed one, unless both the high and the low radiator are connected in series on one pipe, but unless the resistance of the pipes is duly proportioned it may be so serious a hindrance as to prevent the lower radiator getting any circulation at all.

To understand the somewhat complicated interaction of forces in this case, consider the following :—

Flow of Water in Branched Circuits.

If there are two circuits in parallel between the same flow and return main, as shown in Fig. 116, and if in one of them a circulating pressure of

p_1'' of water is generated, and in the other a similar pressure of p_2'', it is evident that the resistance set up by the resulting flow in the branches must be proportional to p_1 and p_2 respectively.

For instance, let the figure represent a branched circuit in which water from the main flow pipe F passes to the main return R through two channels, $F\ A_1\ B_1\ R$, $F\ A_2\ B_2\ R$. Let the mean temperature in column one be t_1, and in column two t_2, t_1 being higher than t_2, and suppose that the pipes $A_1\ A_2$, $B_1\ B_2$, are so large as to have practically no resistance in comparison with the pipes $A_1\ B_1$, $A_2\ B_2$.

Then if the valves V_1 and V_2 are supposed closed for a moment it is evident that the pressure at A_1 must be equal to the pressure at A_2, and the pressure at B_1 will be less than the pressure at B_2.

Suppose the lines $C_1\ D_1$, $C_2\ D_2$ represent the state of static pressure all down the two pipes.

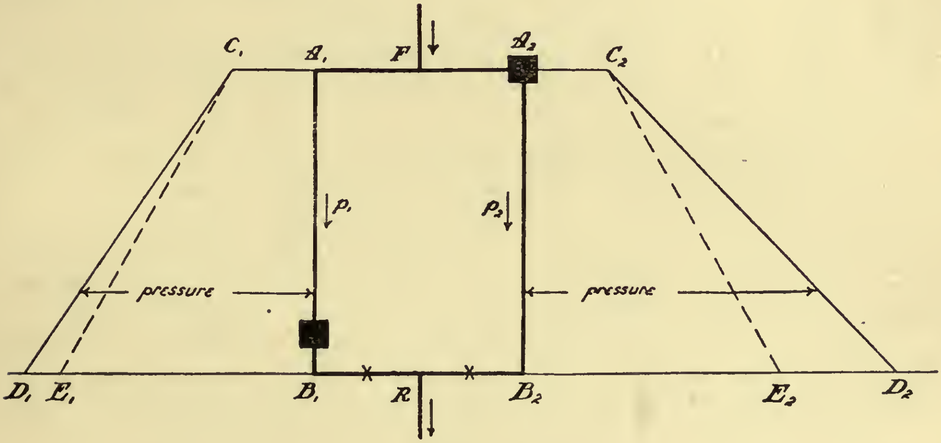


FIG. 116.

When the valves $V_1\ V_2$ are open, and a circulation established, not only must the pressure represented by $A_1\ C_1$ be equal to that represented by $A_2\ C_2$, but also the pressure at the bottom of the column, namely, $B_1\ E_1$ must be equal to $B_2\ E_2$, for it is impossible that two different pressures should co-exist in the same pipe of no resistance. The velocity head is in all practical cases so low that it can be disregarded.

The line of actual pressures when the water is flowing, therefore, will be that shown dotted, namely, $C_1\ E_1$, and $C_2\ E_2$.

It is evident, therefore, that an amount of pressure $E_1\ D_1$, which is the "circulating pressure," must be destroyed by the friction due to the velocity generated in the pipe $A_1\ B_1$, and similarly the amount of pressure $E_2\ D_2$ must be destroyed in the pipe $A_2\ B_2$. In other words, the velocity set up in each pipe must be such as will generate this resistance in each case.

Now it is the object of the design to secure that the sizes of the pipes shall bear such a relation to the required amount of heat, the size of the radiating surface and the available circulating power, that the temperature of the water returning through B_1 B_2 shall be approximately the same in each case, or in other words, the pipe sizes must be so chosen that the quantity of water will be proportional to the required quantity of heat in each case.

Expressed in another way, the general law applicable to any complete circulation, when properly proportioned, is that when the quantity of water necessary to supply all the heat carried by each pipe comprising the circulation is flowing through all the pipes, not only will the total loss of head in the entire circulation from the boiler flow back to the boiler return be equal (as it must of necessity be equal) to the amount of circulating force obtainable from the difference in temperature between the rising column and the falling column of that particular circulation, but also that the water from each circulation will return to the boiler at the same desired temperature.

This general law must be obeyed for every circulation throughout the entire installation, and for every radiator in it. If it is not, it is certain that the radiators concerned will not obtain the proper quantity of heat.

Suppose Fig. 118, on page 475, represents a system of pipes intended to feed a favourably situated radiator at *R* and an unfavourably situated one at R_1. This is a practical example of the previous theory.

In order to investigate the matter let us examine the effect of an alteration in resistance in the various parts of the circuit when the favourably situated radiator *R* alone is connected, as shown in Fig. 117.

As before, the point *D* represents the free water level of the hot rising column when no water is flowing through the system, but the portion of the flow and return columns above *F* and *R* is now assumed to be at the standard temperature of 62°. *E* represents the free water level of the cool return column in the same conditions.

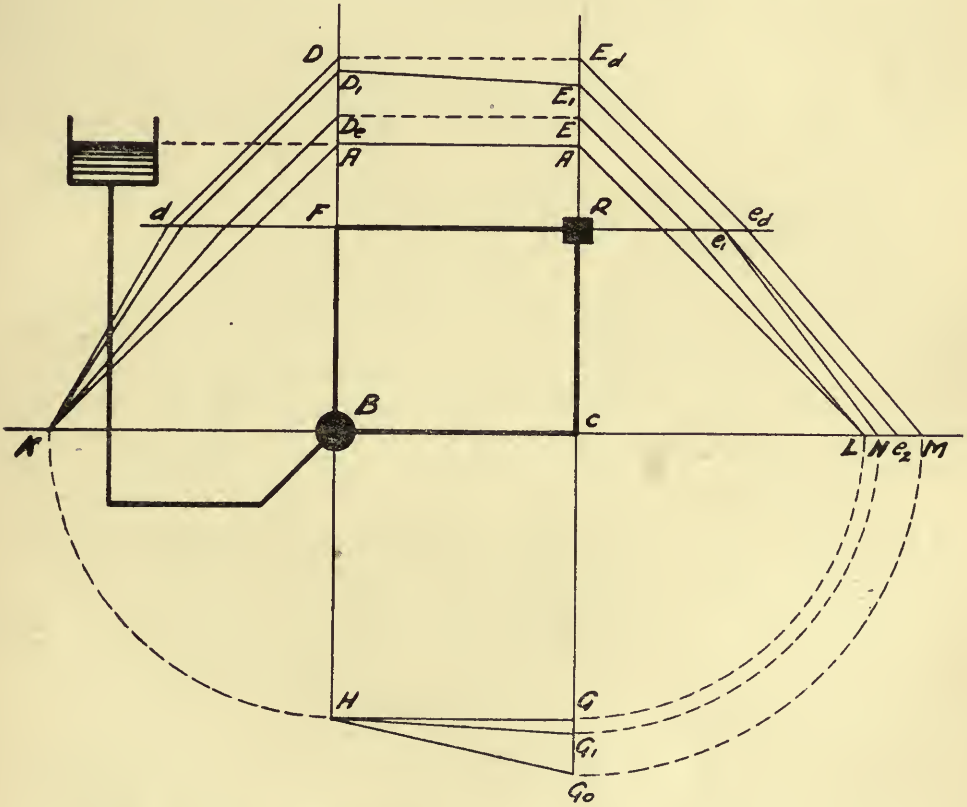
It has been proved in Chapter XX. that the difference in level between *D* and *E* represents the entire circulating pressure which is driving the water through the whole system.

In the whole of the following discussion it is assumed that the temperature of *BF* is always maintained constant at, say, 180°, and of *RC* at 140°, and that a constant quantity Q lbs. per hour of water flows through the system. This implies that the resistance of the entire circuit is maintained constant as a whole, and the changes in resistance of the individual parts described below represent only an alteration in the relative proportion of the resistance.

It has also been explained that the slope away from the vertical of the line representing the pressures in any vertical riser, when the water in the riser is at rest, is greater according as the density in the riser is greater, that is, according as the temperature in the riser is less.

When water is flowing through the riser the friction produces an alteration in the slope of the line away from what may be called the "static line" in such a sense that the absolute pressure becomes less in the direction in which the water is flowing.

It has been pointed out that the effect of allowing the circulation of water to proceed in a circulation of this kind is to reduce slightly the pressure in the upper part of column *BF*. The larger the pipe *BF* is made, that is, the less its resistance, the smaller will be this reduction of pressure,



HOT WATER CIRCULATION.

Diagram illustrating variations in resistance and pressure.

FIG. 117.

until when the pipe *BF* is made so large that its resistance is negligible compared with the resistance of the falling column *RC*, the point D_1 (which, as in Fig. 115, represents the free water level of the hot rising column while the circulation is proceeding) coincides with the point *D*.

The greater the resistance of the pipes *BF* the lower the point D_1 drops below *D*, until when the resistance of the riser *BF* is made very great, compared with the resistance of the pipe *FR*, the point *D* is at D_0 at the same level as *E*.

The pressures right up the rising pipe *BF* are obtained for any given resistance by drawing lines *Dd* D_1 d_1 at 45° and joining the point d_1, wherever D_1 may happen to be, to the point *K*, where the line *BK* represents, as before, the head in feet of water column at 62°, due to the water level in the expansion tank, that is, *BK* = the difference in level between *B* and *AA*.

Now consider the effect of an alteration in the resistance of pipe *FR*. If this pipe is so large as to have practically no resistance, the point E_1 will be raised to *E* at the same level as point D_1, wherever that may happen to be.

The greater the resistance of the pipe *FR* the lower point E_1 falls below D_1, until when the resistance of the pipe *FR* is very great the point E_1 coincides with the point *E*, whatever may be the position of D_1.

Consider now the effect of an alteration in the resistance in the pipe *BC*. When this pipe has practically no resistance, the line of pressures *GH* is practically horizontal and parallel to *CB* at a distance below it equal to *BK*, that is to say, there is practically the same pressure at *C* as there is at *B*, namely, the pressure *BK*. But where considerable resistance is introduced into the pipe *BC*, the point G_1 representing the pressure at *C* falls below *G*.

It will be evident that the lowest possible pressure at *C* is represented by *CG*, or *CL*, or *BK*. This will be the actual pressure at point *C* when *CB* has inappreciable resistance, *i.e.*, when all the resistance of the circuit is concentrated in the pipes *BF, FR*.

On the other hand, the greatest pressure which can exist at the point *C* is CG_0, which exists when point *E* is at the level E_d, and the line e_d *M* slopes at an angle corresponding to the density of the return column *RC* when the water is at rest.

This condition would exist when the resistance of *BC* is very high. *i.e.*, where the resistances of *BF, FR* and *RC* are all practically negligible compared with the resistance of the return pipe. Between these two extremes *EL* and E_d *M* the line of pressures in the falling column *RC* may lie anywhere, according to the proportion of the resistance in the various parts of the circuit.

Now consider the effect of introducing another unfavourably situated radiator R_1 connected to the flow and return mains by a pipe PP_1 in the usual way, as shown in Fig. 118. Assume, for clearness sake, that the resistances of the short pieces of pipe PR and CP_1 are negligible, as indicated by the thickening of these lines.

Assuming the amount of water which flows through pipe PP_1 is not great enough to cause any disturbance of the distribution of pressures previously explained, the direction of flow of water in this pipe depends on two factors—

(1) The difference of circulating pressure at its two ends, P and P_1, produced independently of the pipe itself, and

(2) The density of the water column within the pipe PP_1.

If the sum of these two effects is positive downwards water will flow down this pipe, if negative, water will flow up it, that is, from the return main to the flow main, and if the two contrary influences balance one another, the water in the pipe will be absolutely stagnant.

Assume at first that the temperature in the pipe P_1 P is the same as the flow temperature. In a static condition, therefore, the free level of the pipe P_1 P would be at P_d.

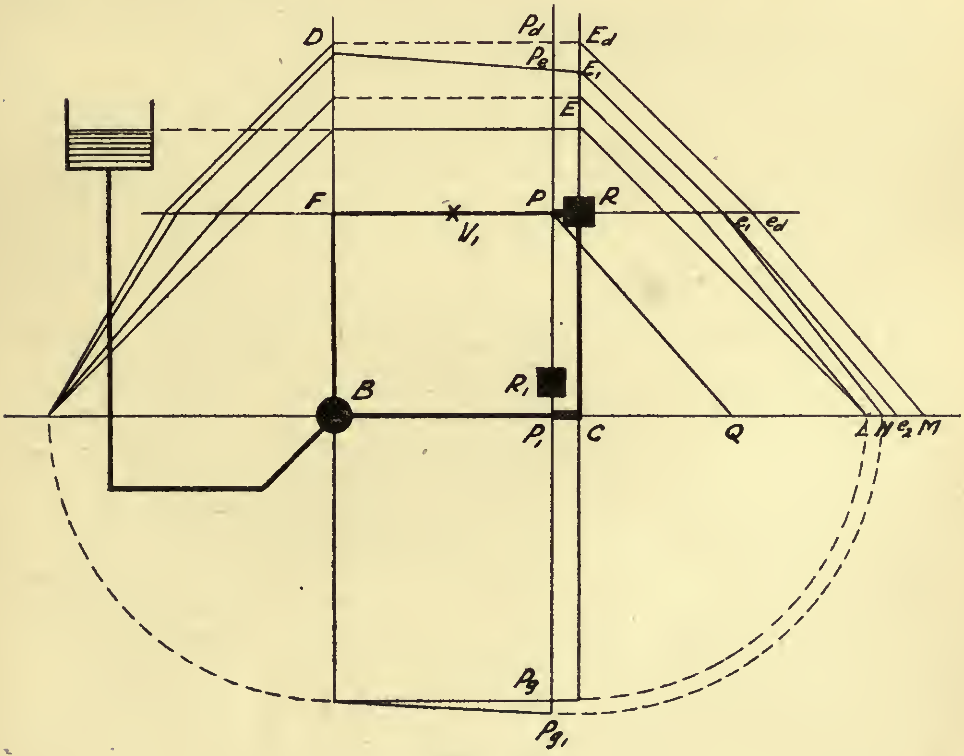


FIG. 118.

In this condition, of course, the circulating pressure at P would be equal to the circulating pressure P_1, and there would in consequence be no flow of water in the pipe if it depended on PP_1 alone. But when water is actually flowing, the absolute pressure at the point P, instead of being PP_d is PP_e, that is, the pressure at the top part of that column is reduced by the amount $P_e\ P_d$.

Similarly at the bottom of the column at P_1 the static pressure would be equal to $P_1\ P_g$, but when the water is flowing the pressure at P_1 becomes $P_1\ P_{g1}$, that is to say, the pressure at the bottom of the column $P_1\ P$ is increased by the amount $P_g\ P_{g1}$.

Thus, there is an increased pressure at the bottom and a decreased pressure at the top of the column $P_1\ P$, and the sum of these two differences of pressure we shall call the "back pressure" tends to drive the water up the column $P_1\ P$ when the flow of water is proceeding in the other part of the circuit *BFRCB*.

It is thus evident that water will certainly not flow down PP_1 unless the excess of weight of the column PP_1 over that of an equal column *BF* is greater than what we have called the "back pressure." The excess of weight must be just so much greater than the back pressure as is required to provide for the resistance of the pipe PP_1.

Now consider the magnitude of the back pressure. It has been shown that the head lost throughout the entire system is equal to EE_d. Of this, the portion $E_1\ E_d$ is consumed in the pipes *BFR*. Since the lines *Ee* are inclined at 45°, $EE_d = LM$, which represents the total circulating power throughout the entire system, where $e_1\ e_2$ is the static line for *RC* with the free level at E_1.

The head lost in the pipe $RC = e_2\ N$, where $e_1\ N$ is the line of actual pressures with the water flowing, and the head lost in pipe $CB = NL$.

It will thus be evident that the back pressure operating against the circulation from P to $P_1 = e_2\ M + LN =$ total head generated throughout the system — head lost in *RC*.

The condition, therefore, for any circulation down the pipe PP_1 is that the excess of weight of the column PP_1 over that of an equal column at the temperature of the riser *BF*, must be at least equal to the difference between the head lost in the whole circuit and that lost in the riser *RC*.

Hence we have the rule that if an *adequate* circulation is to be maintained in pipe PP_1 the total resistance in the pipes *BF*, *FP*, PP_1, $P_1\ B$, when an adequate quantity of water is flowing through each pipe must be the same as the head produced by the excess of weight of the column PP_1 over that of the column *BF*.

In order then that the quantity of water flowing through pipes *BFP* may not be excessive, the *total* resistance of the circuit *BFRCB* must be equal to the circulating pressure generated by the superior weight of the column RC.

In other words, the sizes of pipes $B\ F\ P\ P_1\ B$ must be calculated having regard only to the position of R_1, while the excess of resistance in the circuit $BFRCB$ must be introduced in the pipe RC so as to avoid an excess of water through the pipes BFP, that is, the resistance of RC must be so high as to destroy the additional head due to the relatively great weight of the column RC.

If the circulation in $BFRC$ is too great the relation between the pressures will be such that a condition of stagnation is automatically produced in pipe PP_1, that is, there is a tendency towards stable equilibrium within this pipe.

The condition which ultimately determines whether water will flow down the pipe or not is whether the line of static pressures within the pipe slopes as much as or less than the corresponding line representing the pressures within the pipe RC when the water is flowing, and this, as has been previously shown, depends on the relation between the various resistances.

Assume, for instance, the line of flowing pressures within the pipe RC is represented by some such line as $e_1\ N$. The pressure at the lower part would be considerably higher and the pressure at the upper part is considerably lower than it would be if valve V were closed and the water were at rest.

There is therefore in these circumstances a tendency for water to back up the pipe $P_1\ P$, due to pressure alone. If, however, the line of pressures within the pipe RC were represented by $e_1\ L$, there is no such back pressure.

But if water flows into such a pipe as PP_1 from the pipe FR, the water must, of course, be at 180°, and at this high temperature and low density it does not of itself produce sufficient pressure to force its way down against the difference of pressure produced by the friction of the moving water, as already explained. Indeed, water in this pipe can only force its way down in such circumstances when it is cold, and therefore of no use for heating the radiator R_1.

What would happen in such a case would be that there would be a very slow motion downwards of water within the pipe PP_1, when it is cold enough to overcome the back pressure. The motion must be sufficiently slow to allow the water coming in at a temperature of 180° to cool down until it is nearly cold before it can get up sufficient pressure by reason of its gradually increasing density to force its way downwards against the prevailing difference of pressure, produced by the flow. If the water makes an attempt to flow quicker than this, the temperature in the pipe is at once raised and its mean density falls below the point at which it will move at all. The temperature of the water acts as a kind of automatic brake cutting off the circulating power the moment it becomes effective as a heating medium.

It will be seen, therefore, that the conditions which tend to favour a flow of water through the pipe PP_1 are that the slope of the line e_1 N should be as little inclined to the vertical as possible. The less line e_1 N inclines the better chance has the riser PR_1. The reasons which favour a small inclination for the line e_1 N are small resistances in pipes BF FR, small resistances in the pipe BC, and high resistances in the pipe RC.

The diagram for the case of horizontal underground mains with vertical risers is similar in principle to the foregoing.

Let Fig. 119 represent a system of this kind, having an upper radiator at R_1 and a lower radiator at R_2. Here in this case, if the radiator valve is shut off, the free water level in the flow pipe would rise to D, as before, and the water in the return pipe to E.

The opening of the valve is equivalent to reducing to practically zero the resistance at the top of the circuit, so that the free level in the flow pipe and the return pipe is the same when the circulation is taking place.

Let D_1 be this level. Now if a column of water at the temperature of the flow pipe were at rest at a level at D_1, the line of pressures would be D_1 M. Similarly, if a column of water at the temperature of the return pipe were at rest, the line of pressures would be E_1 L, where DL is greater than QM.

Since the water flows up the flow pipe, the circulating pressure at the bottom of that column must be greater than the pressure at the top, therefore the line of actual pressures will be some such line as D_1 N.

Similarly, in the case of the return pipe, as water is flowing down this pipe, the circulating pressure at the top must be greater than the pressure at the bottom, therefore the line of pressures will be some such line as E_1 S.

Let a radiator be introduced in such a position as R_2, connected to the flow and return columns. It is obvious that the static pressure YY_1 in the return columns at Y is greater than the static pressure XX_1 in the flow column at X, since the density of the flow pipe is less. The difference may be indicated as $(YY_1 - XX_1)$.

Water therefore tends to flow through the radiator from the return pipe to the flow pipe, when no water is flowing. But when the flow is proceeding the back pressure is $(YY_2 - XX_2)$.

If this is positive water tends to flow through the radiator in the wrong direction. The resistances of the respective pipes therefore must be so great that $XX_2 > YY_2$.

If, therefore, the radiator R_2 is to get its due share of water it is clear that the pressure YY_2 must be so much less than the pressure XX_2 that a proper quantity of water will flow through the radiator R_2 impelled by the difference. If it is not the radiator is partly " short circuited."

In other words, the sizes of the horizontal pipe must be calculated so that when a quantity of water is flowing through them, which is sufficient to supply both radiators, the difference of pressure generated by the friction in the horizontal pipes must be so small that it can be overcome by the circulating power generated by the height of the radiator R_2.

The part of the riser above radiator R_2 must be calculated so that when the required quantity of water flows through it, the difference of pressure generated at the level of the second radiator R_2 must be the same as the difference of pressure existing at those points.

I.—Theoretical Value of the Circulating Pressure.

We shall now express in the form of equations the general principles explained above.

(1) Two-pipe system.

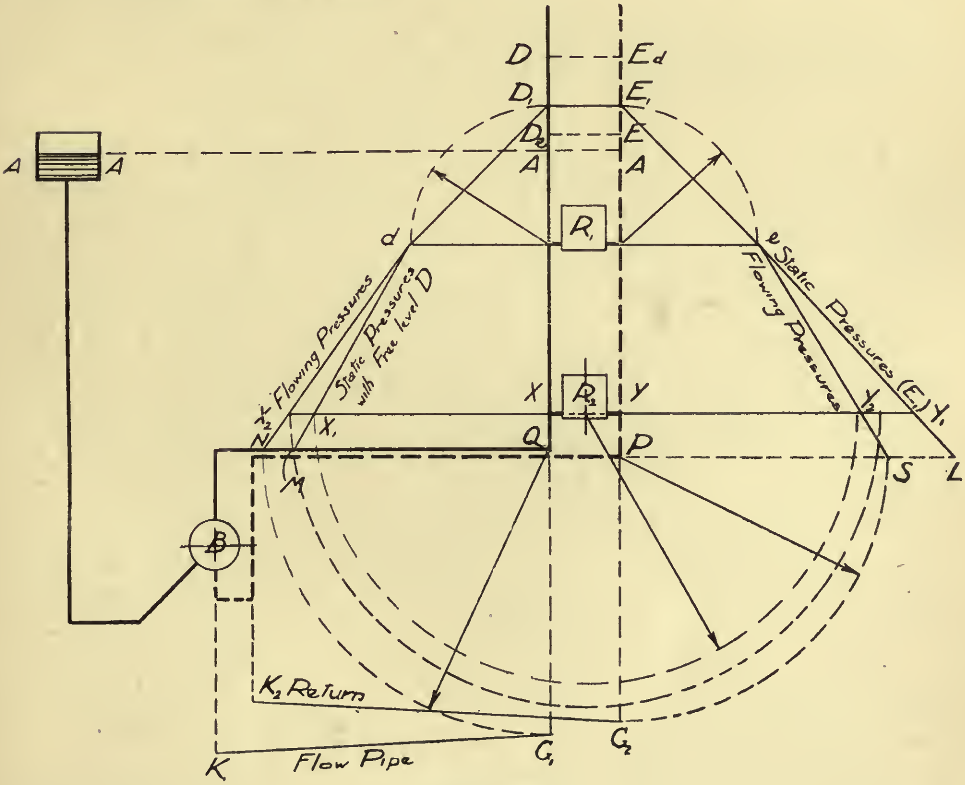


FIG. 119.

Consider first the total circulating pressure produced by a given uniform difference in temperature between flow and return pipes, each having height h, as shown in Fig. 122. In other words, assume a system of pipes which are so well coated that no heat whatever is lost from the system, except at the point R. This represents the simplest possible circulation.

If t_1 and t_2 are respectively the flow and return temperatures, D_1 D_2 the corresponding densities of water in pounds per cubic foot, then the weight of a column H feet high of hot or flow water, one foot in cross section is

$$H\ D_1$$

which also represents the intensity of pressure in pounds weight *per square foot* at the base of the column in which the water is at rest, whatever the size of pipe.

Similarly, the pressure at the base of the cold column at the temperature t_2 in which the water is at rest would be

$$H\ D_2$$

Therefore the difference in pressure or circulating pressure produced by this difference in density is (see last chapter)

$$H\ (D_2 - D_1) \text{ in pounds weight per square foot.}$$

In order to convert this into inches of water pressure, it is necessary to decide what is to be the temperature (and in consequence the density) of the water column or gauge, in which this difference of pressure is expressed.

In order to determine this on a reasonable basis it must be remembered that the formula $\frac{v^2}{2\ g,}$ on which is founded the entire theory of the flow of water in pipes, gives the height of a water column *of the same density as that of the moving fluid.* The height of this water gauge for purposes of using in conjunction with that formula must therefore be calculated *at the same temperature as the flowing water.*

Now if water at any temperature weighs D pounds per cubic ft.

12″ of water column must = D pounds per square foot, or dividing by 12,

1″ of water column = $\frac{D}{12}$ pounds per square foot, or dividing by D,

$\frac{12''}{D}$ of water column = 1 pound per square foot.

Hence, whatever value we choose for D, the circulating pressure in the above case expressed in inches of water gauge at the temperature at which the density is D, is

$$H\,(D_2 - D_1) \times \frac{12}{D}$$

In this case we are dealing with water whose density actually varies between D_1 and D_2, and there is no particular reason why we should select one of these temperatures rather than another, so if we desire approximate accuracy we take for the value of D the mean of the two densities, or

$$\frac{D_1 + D_2}{2}$$

The accurate formula then becomes

Circulating pressure =

$$H\,(D_2 - D_1) \times \frac{12}{\dfrac{D_1 + D_2}{2}} = 24\,H\frac{D_2 - D_1}{D_1 + D_2}$$

inches of water.

This is the most accurate expression that we can obtain. In order that it may be easily used in practice, Fig. 120 and Table XXVIII. have been worked out, showing the value of

$$24\,\frac{D_2 - D_1}{D_1 + D_2}$$

The values in this table when multiplied by the height of the water column in feet, will give in each case the circulating pressure attainable, expressed in inches of water.

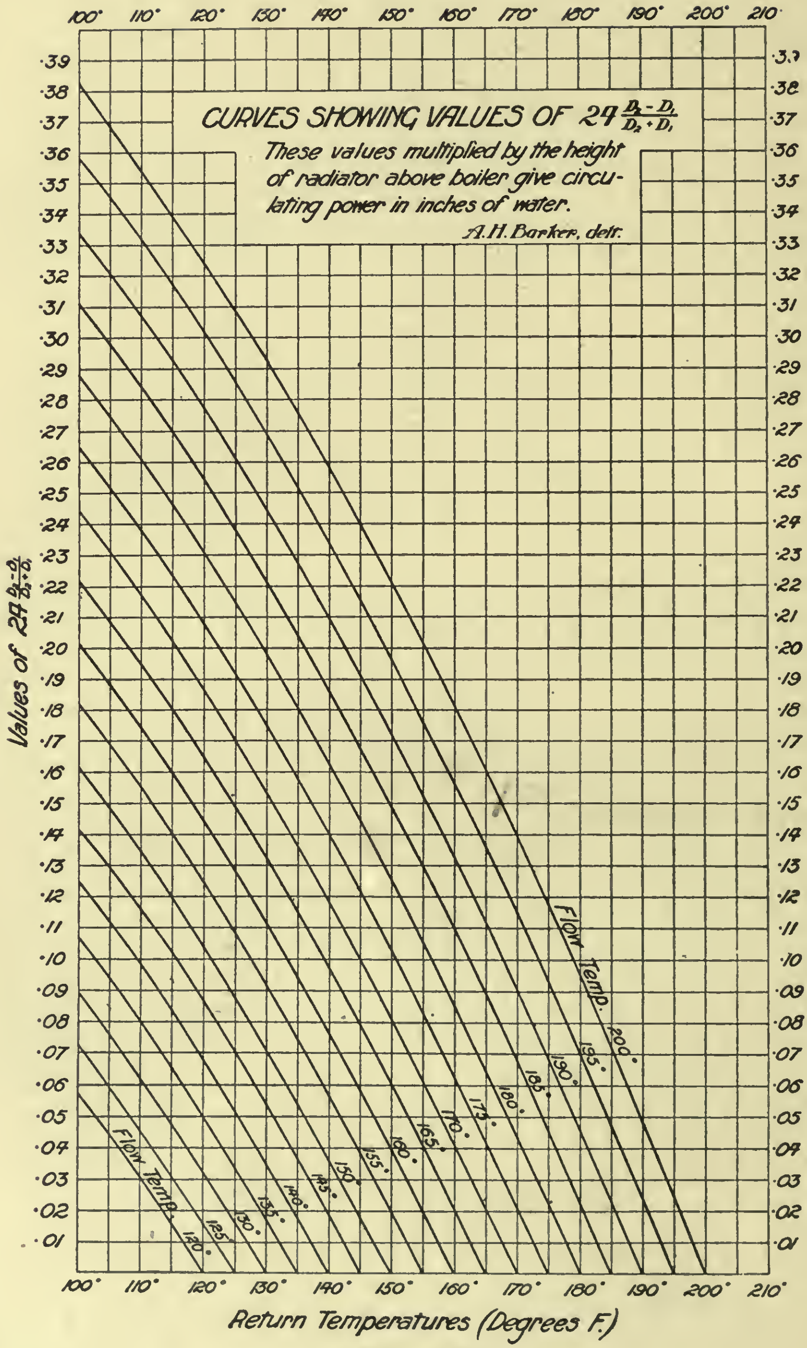
In practice, however, the greatest possible difference of density which will ever occur is between, say, the density at 200°, which is about 60 pounds per cubic foot, and 120° when it is about 61.7—a difference of less than 3%. As this is considerably greater than any actual difference with which we shall be concerned, and at the same time considerably smaller than our probable error in estimating the co-efficient of friction in a pipe, it is clear that an approximation which is near enough for most purposes, would be to take the value of D as 60, which is the actual density in pounds per cubic foot of water at about 200°.

The approximate formula for the inches of water pressure causing flow would then become, for approximate calculations only

$$0.2\,H\,(D_2 - D_1)$$

Circulating Pressures when Losses of Heat from Pipes are allowed for.

It has been previously assumed that there is no loss of heat from the pipes. This is obviously an incorrect assumption, for such a thing as an


CURVES SHOWING VALUES OF $29\frac{D_2 - D_1}{D_2 + D_1}$
These values multiplied by the height of radiator above boiler give circulating power in inches of water.
A.H. Barker, delr.
Values of $29\frac{D_2 - D_1}{D_2 + D_1}$
·01
·02
·03
·04
·05
·06
·07
·08
·09
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·11
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·30
·31
·32
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·34
·35
·36
·37
·38
·39
100°
110°
120°
130°
140°
150°
160°
170°
180°
190°
200°
210°
Flow Temp.
120°
125°
130°
135°
140°
145°
150°
155°
160°
165°
170°
175°
180°
185°
190°
195°
200°
Flow Temp.
Return Temperatures (Degrees F.)

FIG. 120.

absolutely non-conducting coating does not exist in nature. There must in fact be a continuous loss in temperature by every pound of water as it passes along any pipe.

In the case of the circuit shown in Fig. 122 the flow column will not in fact be uniformly at t_1, neither will the return column be uniformly at t_2.

If we had an equation which would give the density of the water at any point in the rising pipe, such as

$$D = f\,(h)$$

then the total pressure due to the rising flow column would be

$$\int_0^H f\,(h)\,dh$$

Similarly, the total pressure due to the falling return column would be, if the density of that column is given by $D = \phi\,(h)$

$$\int_0^H \phi\,(h)\,dh$$

and the circulating pressure would be

$$\int_0^H \phi\,(h)\,dh - \int_0^H f\,(h)\,dh.$$

Such expressions, however, would be far too complicated for practical use, and we must, in practice, assume that the density falls uniformly between the extreme ends of the rising and falling columns.

Thus, if we know that the density of the water leaving the boiler is D_1, and on arriving at the top of the flow column D_2, the most accurate expression we can use in practice is, for the mean density of the rising column

$$\frac{D_1 + D_2}{2}$$

Similarly for the falling return column. We should then get for the circulating pressure

$$H\left(\frac{D_3 + D_4}{2} - \frac{D_1 + D_2}{2}\right) \text{ pounds per square foot.}$$

If, as usual, the flow and return columns consist of several parts, the heights being h_{f1}, h_{f2}, h_{f3}, etc., feet one above the other, then the circulating force in pounds per square foot would be, where D_{f1}, etc., represented the *mean* density in each length

$$(h_{r1}\,D_{r1} + h_{r2}\,D_{r2} +) -$$
$$(h_{f1}\,D_{f1} + h_{f2}\,D_{f2} +)$$

Expressed in words, multiply the height of each section in feet by the mean density of the water in it, in pounds per cubic foot. Add the results together and subtract the sum for the flow column from the sum for the return column. This would give the circulating pressure in pounds per square foot, as accurately as it would be possible to get it in practice.

II.—Circulating Pressure in the Single Pipe System.

The determination of the circulating pressure on the one-pipe system is essentially different from that on the two-pipe system. In the former case a number of radiators are attached to a single pipe, and that one pipe constitutes both the return from one radiator, and the flow for the next succeeding radiators.

Along a pipe, therefore which carries a number of radiators connected to it in this way, there is a continual drop of temperature from the end where it leaves the flow main to the end where it joins the return main.

The pressure produced by such a column depends on the height and the temperature of each of these sections. If the sections are respectively l_1, l_2, l_3, etc., feet long, and are maintained at temperatures t_1, t_2, t_3, respectively, by the action of the radiators at each junction, the densities are D_1, D_2, D_3, etc., the total pressure produced by such a column

$$= h_1 D_1 + h_2 D_2 + h_3 D_3, \text{ etc.,}$$

in pounds weight per square foot. It is to be noted that the temperature of the pipe is to be calculated as constant as far down as the *return pipe* from the radiator, and not, as in the previous case, as far as the centre of the radiator.

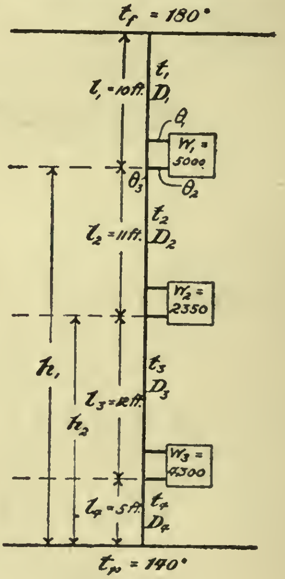


FIG. 121.

The calculations of these temperatures are not so simple as in the case of the two-pipe system.

The principle according to which the calculation may be made is as follows :—

Assume Fig. 121 represents such a drop pipe, and that the figured quantities of heat are taken off from the pipe at the given points, and that the heights are as shown. The total amount of radiation carried by this riser is

$$W_1 + W_2 + W_3 = 11{,}650 \text{ B.T.U.}$$

If, therefore, the difference of temperature of $t_f - t_r$ is decided upon as 40°, the quantity of water which must flow down this riser is

$$\frac{W_1 + W_2 + W_3}{t_f - t_r}$$

or

$$\frac{11{,}650}{40} = 291 \text{ pounds of water per hour.}$$

Whatever the temperature in the respective radiators may be, *assuming that exactly the required quantities of heat are taken off at the corresponding points,* and assuming that the temperature of the upper part of the riser before it reaches the return branch from the radiator is t_f, then that of the second section of the riser will be as follows :—

$$t_f - \cfrac{\dfrac{W_1}{W_1 + W_2 + W_3}}{t_f - t_r}$$

or

$$180° - \frac{5{,}000}{291} = 162.8°$$

This is true whatever the actual outlet temperature of the radiator may be, the only assumption being that the exact quantity of heat, 5,000 B.T.U., is taken off by the upper radiator.

A similar calculation will give the temperature in the third and succeeding sections, as follows :—

$$162.8° - \frac{2{,}350}{291} = 154.7°$$

In practice the actual temperature calculated on for the outlet of the radiator is obtained as follows :—

If θ_1 is the inlet temperature, and θ_2 the outlet temperature, and θ_3 the temperature in the riser on the off side of the outlet, the size of connections and size of radiator are so calculated that

$$3\,\theta_3 = 2\,\theta_2 + \theta_1$$

Thus, in the case of radiator 2 above

$$\theta_1 = 162.8\ \theta_3 = 154.7 \text{ then } \theta_2 = \frac{3 \times 154.7 - 162.8}{2} = 150.6$$

and the mean temperature of the radiator is

$$\frac{162.8 - 150.6}{2} = 156.7$$

Hence, calculating t_1, t_2, t_3, t_4, etc., the temperature in the various sections of the riser, we have the following values :—

$t_1 = t_f$ by assumption $= 180°$ $D_1 = 60.560$
$t_2 = 162.8$ ... ... $D_2 = 60.935$
$t_3 = 154.7$ ... ... $D_3 = 61.104$
$t_4 = 140°$ ... ... $D_4 = 61.388$

Then the actual pressure due to this riser only is

$$10 \times 60.560 + 11 \times 60.935 + 12 \times 61.104 + 5 \times 61.388 - 38 \times 60.560$$
$$= 14.793 \text{ pounds per square foot.}$$

Differences of Temperature between Flow and Return.

It is necessary to decide on the boiler flow temperature, and the return temperature, for which the pipes are to be calculated. In regard to this matter it must be observed that almost all systems of hot water pipes are required, in practice, to work at all temperature, from high to very low.

For a given difference of temperature between flow and return there will be a greater absolute circulating power at a higher temperature than at a low one. Thus, between, say, 180° and 140° the circulating power will be 1.64 inches of water per 10 foot of height, while between 140° and 100° there will be only 1.24″ per 10 foot of height.

On the other hand, the amount of water required to be circulated in order to maintain a given difference of temperature is considerably greater at a high temperature than at a low one (see Fig. 88).

Thus, the quantity of water required per square foot of an ordinary radiator, whose flow temperature is 180° and return 140°, is 3.33 pounds, while that required for the same radiator with flow at 140° and return at 100° is only 1.73 lbs. per square foot per hour. The proportion is 192 : 100

The velocity generated in a pipe (and therefore the amount of water actually circulated) through a given system varies as the square root of the circulating pressure, so that comparing a system working at 180° to 140° with the same system assumed working at 140° to 100°, the proportion of the pressures available would be

$$1.64 : 1.24$$

and the proportion between the quantities of water actually circulated in the two cases would be

$$\sqrt{1.64} : \sqrt{1.24} = 1.28 : 1.11 = 115 : 100$$

It will be thus seen that *in proportion to the water requirements of the circuit with a given temperature difference,* the circulation is more powerful at a lower temperature than at a higher. The result, of course, in practice is that the difference in temperature between flow and return at the lower flow temperature will be less than at the higher temperature.

If, therefore, we work the calculations out for a temperature of 180° to 140° the results will be quite safe for the lower temperature, and further, if the apparatus is correctly proportioned for the higher temperature, it will also be correctly proportioned for the lower temperature.

II.—Theoretical Value of the Friction in the Pipes.

We have dealt in the chapter on flow of fluids with the amount of resistance generated in a pipe of any size by the flow of water at any given velocity v through it. This fundamental equation is

$$p = \rho \frac{l}{d} \times \frac{v^2}{2g} + \Sigma \zeta \frac{v^2}{2g}$$

The result being given in feet of water column, for the explanation of which refer to Chapter V.

It has been explained that the velocity set up will be such that the value of this resistance is equal to the value of the circulating force causing the flow. If the installation consists of one continuous pipe of length l, and of the same diameter d throughout, as shown in Fig. 122, the general equation applicable to the circulation is

$$24\,h\,\frac{D_2 - D_1}{D_1 + D_2} = 12\,\left(\rho\,\frac{l}{d}\,\frac{v^2}{2\,g} + \Sigma\,\zeta\,\frac{v^2}{2\,g}\right|$$

In the case of a compound circuit of pipes perfectly covered so that there is no loss of heat, except at one point in each circulation, precisely the same law holds for every complete circulation into which the whole system can be divided. But in this case, since the complete circulation consists of lengths of pipe of different diameters, having different quantities of water at various velocities flowing through them, we should have for any given circulation :—

$$24\,h\,\frac{D_2 - D_1}{D_1 + D_2} = 12\left(\rho_1\,\frac{l_1}{d_1} \times \frac{v_1^2}{2\,g} + \Sigma\,\zeta_1\,\frac{v_1^2}{2\,g}\right) +$$

$$12\left(\rho_2\,\frac{l_2}{d_2} \times \frac{v_2^2}{2\,g} + \Sigma\,\zeta_2\,\frac{v_2^2}{2\,g}\right) + \ldots\ldots$$

there being as many terms as there are numbers of pipe lengths.

Whether the system is correctly designed or not, the forces of nature will so adjust the flows in various circuits that an equation similar to this must be exactly satisfied for each circulation.

It is the business of the engineer so to determine the sizes and position of the pipes and radiators, that the natural forces acting in this way *will* cause the desired quantity of water to flow through each circuit, and in consequence maintain the desired temperature at the various points.

The Equation of the Circulation.

Assuming that we know the velocity of the water in every length of pipe in the entire system, then any stretch of pipe whose length is l_1, and diameter d_1 in feet, we have

Pressure lost by friction in this length =

$$\rho_1\,\frac{l_1}{d_1}\,\frac{v_1^2}{2\,g} + \Sigma\,\zeta_1\,\frac{v_1^2}{2\,g} \text{ in feet of water column,}$$

or

$$D\left(\rho_1 \frac{l_1}{d_1} \frac{v_1^2}{2g} + \Sigma \zeta_1 \frac{v_1^2}{2g}\right) \text{ in lbs. per square foot.}$$

A similar equation would hold for every length in the entire circulation, and the total resistance would be their sum, so that the accurate expression for any circulation would be

$$(H_{r1} D_{r1} + H_{r2} D_{r2} + \ldots .) - H_{f1} D_{f1} + H_{f2} D_{f2} + \ldots .)$$
$$= D_1 \left(\rho_1 \frac{l_1}{d_1} \frac{v_1^2}{2g} + \Sigma \zeta \frac{v_1^2}{2g}\right) + D_2 \left(\rho_2 \frac{l_2}{d_2} \frac{v_2^2}{2g} + \Sigma \zeta \frac{v_2^2}{2g}\right) + \ldots .$$

CHAPTER XXIV.

DETERMINATION OF PIPE SIZES.

II.—SIMPLIFICATION OF CALCULATIONS IN PRACTICE.

THE scientific principles on which the calculations are based having been explained, it is now necessary to examine the application of those principles to the conditions of practical work, and to give particulars of simplifications and adjustments of the calculations which may be adopted in practice in order to save time.

The Heights of the Flow and Return Columns.

When hot water passes into a radiator it is cooled by the natural loss of heat from the surface, and leaves through the return connections at a lower temperature than that at which it entered. The water within the radiator at different points has different temperatures, ranging at least from that of the flow to that of the return. It has been explained in the chapters on loss of heat by radiation that the practical range of temperatures will even exceed this. There will be parts of the radiator which are actually at a lower temperature than the return pipe.

The boiler, like the radiator, will contain water at all temperatures between t_f and t_r.

The circulating power depends on the difference in weight of the *entire flow and return columns,* including the radiator and boiler. The precise distribution of temperature in the boiler and the radiators is impossible to determine—for it depends on a large number of factors which cannot possibly be taken account of. It is therefore impossible to obtain an expression which gives correctly the whole weights of the water columns, including that in the boiler and radiators.

The assumption is therefore made—and it is found to be sufficiently near to accuracy for practical purposes—that the whole radiator is concentrated on the horizontal centre line, midway between the levels of the flow and return connections. The temperature of the water in the radiator is assumed to be the same as that in the flow pipe as far up or down as the centre of the radiator, below which the temperature is taken as that of the return pipe.

Similarly, the horizontal centre line of the boiler midway between the flow and return connections is taken as the bottom of the flow column, or in other words, the upper half of the boiler is calculated as part of the flow column, and the lower half as part of the return. All calculations are made between the centre lines of the boiler and radiator.

That this assumption cannot be far from actual accuracy is obvious, but note must be taken of it and its reasons.

Temperature in Pipes.

A further approximation which is commonly used in determining the circulating power is to assume that the water maintains the full boiler flow temperature as far as the centre of the radiator, and below that level the water is assumed to have the temperature of the boiler return. The power calculated by this method, though it is only a somewhat rough approximation, will not in ordinary cases be far from the actual power, as will be evident from the following considerations.

With Overhead Flow Main.

Where the flow pipe is above the level of the radiator, the falling part of the *flow* pipe is in reality at a lower temperature than the rising part immediately above the boiler.

The circulating power calculated on the assumption that the flow pipe maintains its temperature as far as the radiator will therefore be *less* than the actual circulating power, for the cooling of the pipe produces a circulating pressure of its own, independently of that produced by the radiator.

On the other hand, the assumption that the falling part of the return pipe from the radiator into the return main is at the same temperature as that at which it enters the boiler (or in other words, that this portion of the pipe is at a lower temperature than it really is) will have a contrary tendency, namely, to make the calculated circulating power *greater* than the actual.

The assumption made in this approximation, therefore, is that these contrary influences balance one another.

It is evident, however, that the drop in temperature between the rising and falling parts of the flow pipe will generally be greater than that between the falling part of the return pipe and the part where it enters the boiler, and that therefore the increase in power due to that difference in the flow pipe will generally be greater than the decrease due to the same circumstance in the case of the return pipe.

The loss of heat is greater and the density varies more rapidly per degree difference at the higher than at the lower temperatures. On the whole, therefore, this assumption is on the safe side. When the radiator is situated near the middle of the falling pipe, the actual circulating power will be greater than that calculated on this assumption, but only to a relatively slight extent.

It is found to be sufficient in practice to omit the exact calculation altogether for overhead systems, and to assume that the circulating power is obtained from the total difference of temperature ($t_t — t_r$) at the height of the radiator.

There are cases, however, in which all or many of the radiators are at or below the level of the boiler. In these cases the whole of the circulating power is derived from the cooling effect on the overhead pipe.

If it were not for this cooling there would be no circulating power at all, and in these cases the complete calculation must be made.

Except in special cases such as this it is found that in practice the general omission of the calculation of the drop of temperature in the mains leads to good results, especially with the overhead system. The amount of surplus power actually obtained above the calculated value, as explained above, provides just about the amount of surplus required in practice to provide for such effects as the interior roughness of pipes, which cannot possibly be calculated for.

With Underground Flow Main.

In the case of the two-pipe system with underground mains the loss of heat from the flow pipe diminishes the circulating power instead of increasing it, as in the case of the drop system. The temperature of the water in the rising flow pipe is less than the boiler flow temperature. At the same time and in consequence, the difference of temperature between the flow riser and the return riser is less than it would be if the flow riser were hotter. Hence, on the whole this assumption gives a circulating power greater than that actually obtained.

This is the real reason why the drop system is so much more energetic in circulating power than is the underground system.

When the total circulating power obtained from or destroyed by the cooling of the pipes does not exceed 5% of the total power it is generally safe to omit the exact calculation. In cases where the boiler is in the centre of the installation the omission affects all the radiators in the system to an approximately equal degree, and therefore the effect is approximately equivalent to a reduction or increase (as the case may be) in the resistance of all pipes throughout the installation, or, on the other hand, to a greater or less difference of temperature between flow and return.

Although with the drop system the flow of water tends to exceed the calculated quantity, and with the underground system to fall short of it, the effect is distributed more or less uniformly throughout the entire installation, and does not disturb the general balance of the system, which latter is the really important consideration.

In cases, however, where an installation is composed of two or more parts on different systems of circulation, such, for instance, as when half of an important installation is on the overhead system and half on the underground system, or in cases in which the boiler is at one end of a long or widely extended installation, it is very desirable to make the complete calculation, and not to rely on this approximation.

With these two assumptions in mind, consider the small installation shown in Fig. 122. Here the force producing circulation is the excess in weight of the column *KC* over that of *LF*.

The portion of the rising column *GF* is equal to that of the falling column *KR*, and as the waters in these two lengths are by assumption at the same temperature, they have the same density, and they exercise no influence in producing movement in the circulation. Below *R*, however, all the water is at the return temperature.

Similarly, below the level of the centre of the boiler *B* the portion of the circuit *MC* balances the portion *LB*, as by assumption these two portions are at the same temperature.

Thus, it will be seen that on the above assumptions the operative parts of the column which really produce the movement consist of the hot column *BG* at the boiler flow temperature balanced against the cold column *RM* at the temperature of the return. In other words, *the height of the column whose expansion causes the circulation, is the height between the centre of the boiler and the centre of the radiator in question.*

Single Pipe System.

In the case of the single pipe system the accurate method of determining the circulating power due to the risers has been explained on page 484.

This is somewhat cumbersome for an approximate calculation.

It is possible to obtain an approximation which is sufficiently near for purposes of making preliminary calculations, as follows :—

Find the position of a point which is at the centre of gravity of three weights bearing the same relation to one another as the respective quantities of heat required, and situated respectively at the level of the return connections of the corresponding radiator. Assume that the riser has the flow temperature as far down as the point so found, and the return temperature below it. The circulating power obtained by this assumption will not differ greatly from that obtained by the accurate process (see Fig. 123).

The method of finding the position of this point is as follows :—

Let h_1 be the height in feet of the return connection to the top radiator, which has an emission of w_1 B.T.U. per hour. h_2 w_2, h_3 w_3 being the corresponding values for the lower radiators, then H_r the required height, is obtained from the following equation :—

$$H_r = \frac{h_1 w_1 + h_2 w_2 + h_3 w_3}{w_1 + w_2 + w_3}$$

In order to illustrate that the two methods of calculation give approximately the same result, take the example shown in the figure.

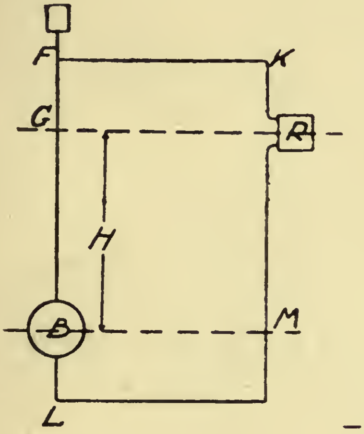


Fig 122

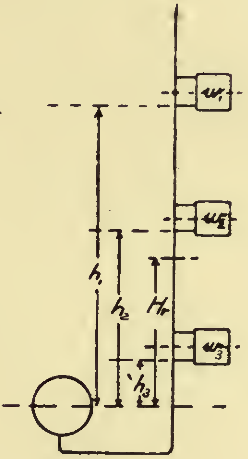


Fig 123

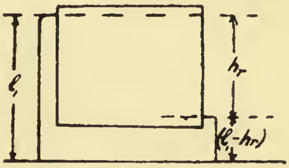


Fig 124

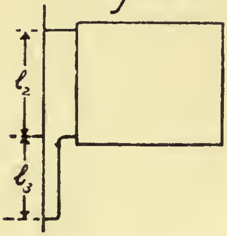


Fig 125

The correct calculation is given on page 485, and the result is 14.793 lbs. per square foot. The approximate calculation is as follows :—

Height of centre of gravity above return pipe =

$$\frac{5{,}000 \times 28 + 17 \times 2{,}350 + 5 \times 4{,}380}{11{,}650} = 17.3 \text{ feet.}$$

Circulating pressure due to a column consisting of 20.7 feet at 180° and 17.3 feet at 140°

$$= 17.3\ (61.388 - 60.560) = 14.05 \text{ lbs. per sq. foot.}$$

The other calculations for the single pipe system are the same as for the two-pipe system.

In cases where radiators are attached to a horizontal pipe on the single pipe system, the principle on which sizes of connections may be determined is as follows (see Fig. 124):—

The actual diameter adopted is not of itself of any considerable importance provided the radiator is of the correct size to give off the required amount of heat. The smaller the size of pipe used the lower will be the mean temperature in the radiator, and the larger it must be. But provided the exact quantity of heat is taken off the size of the pipe has no influence on the temperature in the main.

If the full difference of temperature between the flow and return pipe be assumed, the rising connection to the radiator will have a length l_1 and a temperature t_1. The pressure due to this will be $l_1 D_1$.

The falling column must be assumed to consist of two parts—

(1) The height of the radiator h_r between centres at the mean density of water in the radiator, $\frac{D_1 + D_2}{2}$

(2) The length of the return, $l_1 - h_r$ at the temperature t_2.

The total circulating power is therefore

$$h_r \left(\frac{D_1 + D_2}{2} \right) + (l_1 - h_r) D_2 - l D_1$$

EXAMPLE.—Let the radiator be 50 sq. ft., $K = 1.23$, and difference of temperature = 83

$h_r = 2.16$

$l_1 = 3.66$

$$\text{Circulating pressure} = 2.16 \times 61.331 + 1.5 \times 61.655 - 3.66 \times 60.998$$
$$= 1.68 \text{ lbs. per sq. ft.} = 0.32'' \text{ of water.}$$

Here the quantity of water required =

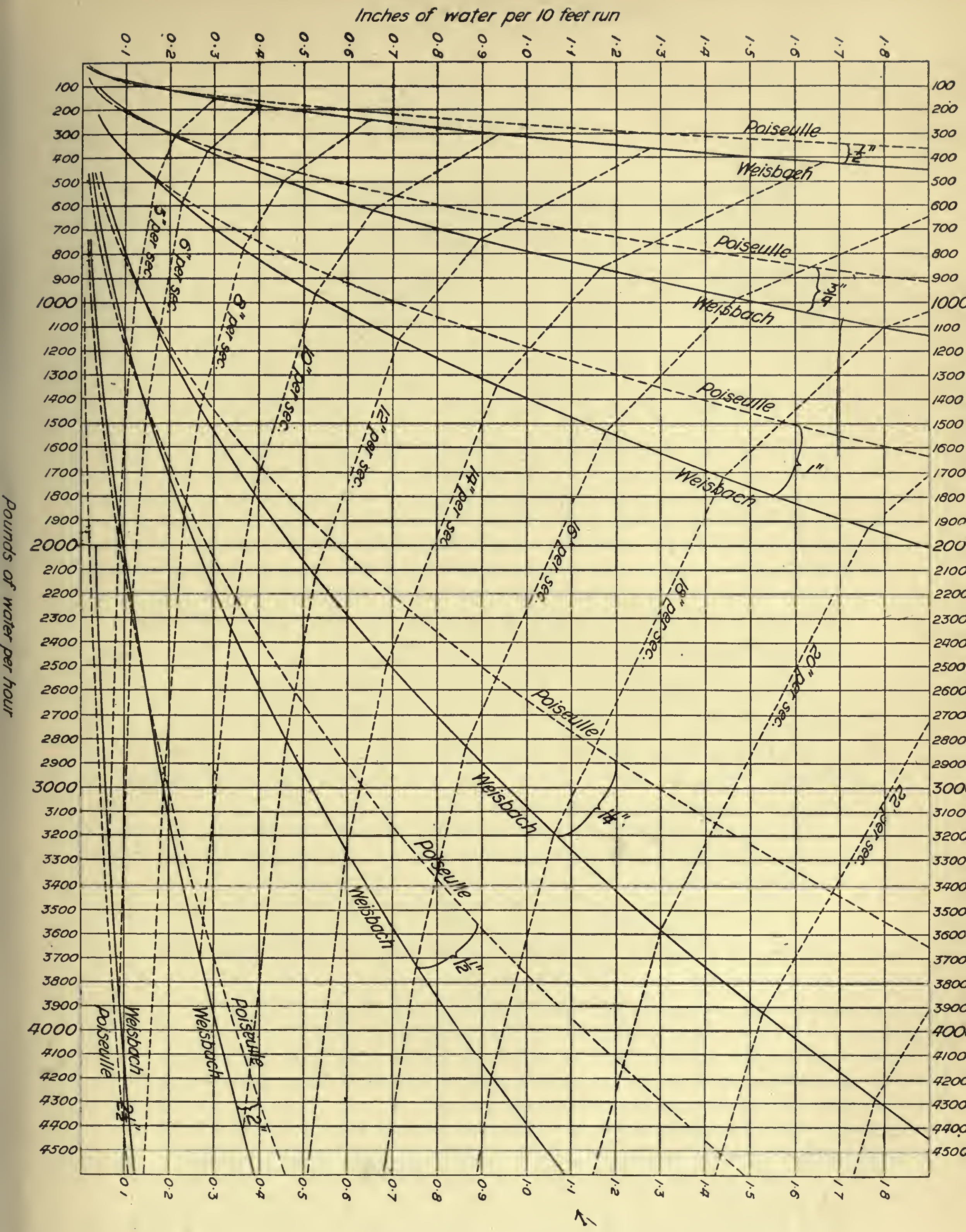
$$\frac{50 \times 1.23 \times 83}{35} = 146 \text{ lbs. per hour.}$$

The equivalent length of pipe may be generally taken as about 10 feet including single resistances. The tables show that a head of 0.32″ of water will force 360 lbs. of water per hour through 10 feet of ¾″ pipe, and with ½″ pipe only 124 lbs.

If the diameter is larger than this the temperature difference will not be so great, and if smaller the difference will be greater.

If it is desired to increase to any reasonable extent the circulating power to a radiator fixed to a vertical pipe on the drop system, it can be done in the manner shown in Fig. 125.

By the adoption of this device the circulating pressure through the individual radiator will be increased, but that through the riser as a whole will be reduced as it is equivalent in effect to lowering the position of the radiator on the riser.



FRICTION OF WATER IN PIPES

Comparison of formulae of Weisbach and Poiseulle

A. H. Barker, delt.

FIG. 126.

A formula which determines a satisfactory size of connection to a radiator in a general case on the single pipe system is

$$d = .01\sqrt{\text{B.T.U.}}$$

where d is the internal diameter of the pipe in inches.

This formula gives the following sizes of pipe connections to radiators on the single pipe system.

Dia. of pipe.	B.T.U.	Approx. size of Radiators at 140 B.T.U. per sq. ft.
½″	2,500	18
¾″	5,640	40
1″	10,000	72
1¼″	15,600	112
1½″	22,500	160

Assuming a drop in temperature in each case of 40° F. this range of sizes corresponds to a velocity of water through the pipe of 2.55″ per second.

Practical Considerations Relative to the Flow of Water in Pipes.

It has been pointed out that there are in existence a considerable number of different formulæ evolved by competent experimenters, which purport to give the amount of water flowing through a pipe of any diameter and length under the influence of a definite head.

These formulæ vary very considerably among themselves for reasons which are not difficult to understand.

Two such formulæ representing results of the work of two experimenters have been explained and compared in the chapter relating to flow of fluids. Fig. 126 shows a graphical comparison between these.

It is only necessary to consider the conditions under which all such tests are necessarily made, and to compare those conditions with those under which a heating apparatus works, to understand not only the limitations of the application of such formulæ to practical work, but also the wide difference between the result of the work of different experimenters.

It would be obviously useless for any experimenter to make any exact tests on pipes which were not fairly clean and smooth, for it would be impossible to obtain any definite and consistent results unless the pipe experimented on were in that condition.

Now the pipes in use in a practical heating apparatus are frequently neither clean nor smooth.

The nature of the frictional forces in the pipes are the tendency of the water to adhere to the sides of the pipe, and to resist the deformation in its own substance. If the interior of the pipe were a perfectly smooth cylinder, it is conceivable that very uniform and exact results might be obtained by different experimenters, giving the exact amount of water flowing through a pipe under certain exactly specified conditions.

The amount of water which would flow (under a given head) through any commercial pipe of any nominal size, necessarily varies very greatly, not only with differences of diameter, which are always found in different lengths, but also with the nature of the interior surface with which the water is in contact.

As the substance of which the great majority of pipes used by the heating engineer is made, is such that by corrosion and pitting and other causes, the condition of the interior of the pipe is continually altering from week to week, it appears impossible ever to obtain a formula which will give, with any degree of scientific exactitude, the amount of water which will flow through a commercial size of pipe in any given conditions.

It is clear, therefore, that the amount of water which a pipe will deliver is a question, not only for practical experiment, but also for subsequent practical experience. So far as it concerns the heating engineer, the root of the whole question is this:—

"If in a certain installation pipes of a certain quality and of a certain size are put in, and water is allowed to pass through them for, say, one or two years, what size of pipe will at the end of that time pass the quantity of water which is required, having in view the fact that the interior of the pipes will at that time have become rusty, corroded and pitted, and very considerably altered in surface, and to some extent perhaps in size and shape from what they were when put in?"

The main point is that the quantity of water flowing through each pipe should be *adequate*. It is of far less importance that the pipes should be as small as possible. It is therefore clear that whatever formula is adopted for determining the sizes of pipes, if it is to be satisfactory in working, it must give results which, when the pipes are new, may be considerably in excess of what is required for the most favourable conditions.

Different brands of iron and steel vary considerably in their behaviour when subject to rust and corrosion for a long period. It is, of course, impossible in practice to have different formulæ corresponding to different materials. The quality of water, too, which is employed in different installations varies very greatly indeed. Any system of pipe sizing therefor, howsoever scientific and accurate it may be when the pipes are new, is a bad system unless it allows for this very practical question of the alteration in the interior condition of the pipes.

Fortunately for the heating engineer, however, in most cases the alteration which does actually take place in the interior of a pipe is in the direction of allowing freer flow when the pipes are old than when they are new, but not in all cases, and not for pipes in all positions.

The general effect produced by the prolonged use of pipes for circulating water is naturally a rusting of the interior of the pipe which gradually eats the pipe away. This, however, does not *necessarily* imply that the pipes will pass more water. A perfectly smooth pipe, lined, say, with glass, would pass more water under the same head than would a slightly larger pipe whose interior surface was rough and irregular. In practice we are concerned with pipes as we find them, and any satisfactory figures must be suitable for application to rough ordinary pipes which have actually been used for years in an installation.

Even this, however, does not necessarily imply that the results are entirely reliable in all circumstances, for rust or scale which is formed on the inside surface of a pipe ultimately breaks away from the pipe and is sometimes carried along the pipe and deposited wherever it finds a resting place. If such accumulations of rust and scale are sufficiently great, partly to block up the pipe, no possible calculation can allow for it.

It is found by experience that the pipes most liable to become blocked in this way are bends and elbows in the lower pipes in a system or installation, such as the pipes of the ground floor or basement, and particularly the return pipes. In some of the rapid circulation systems introduced during recent years, which work with very small pipes, this effect is particularly marked. The effect may even be unnoticed during the first season in which the apparatus is at work. The circulating power may be so great as to keep such deposit from accumulating to any great extent, by keeping the solid matter in suspension or circulation in the pipes along with the water. When, however, the installation is shut down for the summer season all such deposit or sediment gradually sinks to the lowest position that it can reach and settles there. In the course of the four or five summer months in which it is left at rest it sometimes partly hardens into a mass which the subsequent efforts of the circulator cannot move, and forms a partial or complete block in the pipe.

The only way of getting rid of this trouble is to supply at all points where such sediment is likely to accumulate, something in the nature of a small dirt trap or to arrange the connections in such a way that they form a chamber of a size that will accommodate or contain the sediment, leaving a sufficient area for the water to pass in spite of its presence. These traps may be cleaned out periodically when required, but it is not, in practice, found necessary to clean them out frequently, after the dirt and debris, always found in a new installation, is removed. It is generally necessary to examine such installations about once in four or five years.

In practice the upper pipes of a system are generally left absolutely clear of deposit, even although the pipes may have been considerably corroded by water.

As a hot water heating apparatus is generally kept entirely full of water, the corrosion or rusting is nothing like so great as might be expected, even after prolonged use, 20 years or more.

It appears necessary for the formation of rust, either that air or some acid should exist in the water. If such conditions are entirely absent, the rusting is small, and almost negligible over a long term of years.

The term "corrosion" has been and is frequently incorrectly applied. Corrosion strictly applies to the eating away of the substance of a pipe by corrosive or chemical action of the water on the material of which a pipe is made. It has been and is often incorrectly applied also to the process whereby the dissolved solids in the water are deposited in the interior surface of the pipe by heat or by evaporation of the dissolving water. It is clear that the term "corrosion" used in this sense must in all cases diminish the flow of water through a pipe. In this case also no conceivable calculations can allow for the effect of cumulative deposits of unknown and unknowable magnitude. If this kind of deposit continues it is simply a question of time for a pipe of any size to become absolutely or partially blocked and so to stop entirely or partially all flow of water through it.

Considerations on the Choice of a Formula.

Whichever formula is adopted as a working hypothesis, the corresponding nominal capacity of a pipe of given size will vary accordingly.

It is very unsatisfactory for the engineer to be obliged to decide which of these widely differing formulæ he will use. The essential danger attending the use of any formula whatever will be understood from a consideration of the following:—

Every formula expressing the resistance to flow of water in pipes is (or should be) nothing more than a condensation into one expression of the results of a large number of experiments made on different sizes of pipes of accurately determined diameter, and under different conditions.

The use of the formula involves the assumption that all the results obtained *can* be and have in fact been condensed into one expression, and that any variation from accuracy involved in its use will not be sufficiently great to affect the results in practice.

If the formula is to be of use in the accurate solution of practical problems, it is necessary that it should take account of all possible conditions and circumstances which may have a practical bearing on the result. The acceptance of a formula implies the assumption that every factor which has a bearing on the result which is not taken account of by the formula, either has no effect or has so small an effect that it may be neglected.

If, therefore, it is possible to point to one variable set of conditions or one variable factor which has an *important* influence on the result, and which the formula takes no account of, that alone is proof that for practical purposes the formula is not reliable.

It is evident, too, that the very effort on the part of the experimenter to compress an immense number of experimental results into one formula, constitutes a danger to those who have to use the formula. This effort must of necessity involve compromise, correction, adjustment, and other processes, which, however necessary they may be to make theory and experiment agree, cannot fail to be looked upon askance by the man who puts his money on the result. The latter requires the correct results of careful experiments, not the proceeds of mathematical agility.

It might, for instance, be found that a particular formula would hold very well for medium sizes of pipe, but be widely in error for very small or very large diameters. If either were found to be the case there would be an obvious temptation to disregard or compromise for the peculiarity, rather than to recast the entire formula so as to take account of it, a process which might very easily complicate the formula very much. The formula might, in practice, be employed for sizes much smaller or much larger than those on which the experiments were made.

A formula is only necessary or desirable in cases where the number of different possible sizes is so great that individual experiments on each size are impossible or impracticable. This is not the case with the water pipes used by the engineer.

But there is a further and yet more important fundamental objection to the employment of any formula whatever for such a purpose as this. If the formula is even tolerably accurate from a scientific point of view, as already stated, it must refer to smooth pipes of accurately determined diameter.

The interior roughness of the pipes does in fact make not only a difference, but a large difference in the friction. It is a fundamental necessity, therefore, that if a formula is to apply accurately to pipes of different degrees of roughness, it must contain some factor relating to the degree of roughness in the pipes.

If, as in the Weisbach formula, which is in common use on the Continent, it contains no such factor, it must be assumed that the coefficient is unity, or, in other words, that the pipes experimented on are to be assumed smooth internally, and also that the diameters are accurately expressed.

It is also well known that changes in the temperature of the water do produce changes in its viscosity (and therefore in the friction) as well as changes in density. Since the formula of Weisbach takes no account of the change of viscosity or temperature except so far as it affects the density, it must be to a certain extent incorrect on this account alone—to what extent will be made clear by an examination of the formula of Poiseulle—assuming it to be accurate.

It is probable that Weisbach's formula is fairly correct when applied to smooth pipes of an accurate diameter, with water at a certain temperature and with the surrounding air at the same temperature. But even if this is so, much more information is required before the formula can be safely accepted for the purposes of the heating engineer.

The latter is not concerned with internally smooth pipes of accurately determined diameter, but with a definite range of sizes of commercial quality of pipes of different *nominal* diameters—a very different proposition indeed. Not only has he no guarantee as to the internal smoothness, but he does not know even that the actual internal diameter even of clean pipes will be the same as the nominal diameters. Indeed, if he has ever measured them, he knows quite well that they are not.

The Engineer is in fact concerned with no less than four different qualities of wrought iron pipes, known as the hydraulic, steam, water, and gas qualities, all of which for the same nominal diameter, differ in actual internal diameter.

The process, therefore, of calculating the friction from the *nominal* diameter by means of a formula determined on scientific principles, involves closing one's eyes to the *realities* of the problem, treating it in fact as a problem in mathematics instead of a problem in engineering.

It is no uncommon experience for those who are testing the actual friction in such pipes to find a difference of 10, 15, 20, or even more, per cent., between the friction in two pipes of the same apparent diameter, due to differences in the perfection of manufacture. Neither is it at all infrequent to find differences of 10% in the *actual* diameter between two pipes of the same nominal quality and diameter. In fact, as applied to the practical problem, it appears to the present writer that it is very undesirable to turn the results of experiments into a formula at all, except for the purpose of explaining the scientific principles on which the theory is based.

The values used in practice, according to this view, should be the results of careful and direct experiments on actual pipes of all the standard sizes of the same or similar quality to those which it is intended to use, and used at approximately the same temperature and fixed in a similar way.

The conditions are entirely different in the case of flues used for the passage of air. Such flues are made of all diameters and of all sizes. They are sometimes rectangular, with any proportion between width and depth, and often of quite irregular shapes, but generally of fairly uniform quality of surface, and the size is generally correct, and there is no such thing as deposit in them.

For air flues, therefore, a formula is not only a *necessity* in order that it may be possible to calculate the result for any particular size or shape, but the actual friction is more likely to correspond with the results of experiments.

A formula is also perhaps more desirable for the Continental engineer than for the British for determining the flow of water. The former uses a much wider range of sizes than does the latter, owing partly to the fact that pipe diameters are measured on the Continent in millimetres, and in England in inches—a much larger unit—and owing also to the fact that the pipes used on the Continent are probably on the whole smoother and more accurate in diameter than those found here.

Roughly speaking, on the Continent the standard sizes of pipes commonly used are the same as in this country up to 1½″ diameter, although these standard sizes vary by one or two millimetres. Above that diameter, the different standard sizes abroad are much more numerous than with us, so that independent experiments on each standard size would be much more cumbersome than in this country.

Extensive experiments on this matter by the Author, which, however, are not complete at the time of writing, show clearly that the capacities of given nominal sizes as manufactured in this country do not agree with the Weisbach formula, either with hot water or cold, and in the case of the smaller sizes of wrought iron pipes the discrepancies are serious. The larger sizes of pipe agree with the Weisbach formula much more nearly than the smaller.

For English practice it would be better, according to the view of the Author, to base the sizes on experiments made on actual pipes of different *nominal* sizes, as supplied by British makers such as are in common use in British installations. Unfortunately the series of such experimental results are not up to the present completed. The tables in this work are therefore prepared from the Weisbach formula, which, as explained below, even though they are considerably in error, are at least self consistent, and if correctly employed in calculations do in fact secure a good balance in a system of pipes. This is of greater importance than exact accuracy.

As mentioned above, there are four commercial qualities of such pipes, known as the hydraulic, steam, water, and gas quality. Of these the hydraulic quality is not used on any except high pressure hot water installations.

The steam quality is often, though by no means always, used in first class installations in this country. Even the "gas" quality is sometimes used in cheaper installations.

In the case of the latter two qualities the substance of the material is thinner, while the outside diameter remains the same as the steam quality, in order that the same dies and screw threads may be employed for all qualities. The lower the quality of the pipe adopted, therefore, the greater in ordinary circumstances will be the amount of water circulated under a given head.

General Consideration of Rules for Pipe Sizing.

It has been before explained that the object of any correct system of pipe sizing is twofold, not only to ensure that each pipe shall be large enough to supply the required quantity of heat under the existing circulating head, but also that any particular pipe should not be too large so as to return its water at too high a temperature.

It is shown in the last chapter that if in one part of a system the pipes are too large, the result will be that pipes in other parts of the system may thereby be rendered too small.

It is impossible that any particular series of experiments should apply accurately to all pipes, however fixed. Strictly speaking, they apply accurately only to pipes of exactly the same quality, fixed in the same way, and worked under the same conditions, as those on which the experiments were made.

For practical purposes it is equally impossible to have a large number of different experiments each applying to pipes of different qualities and worked under different conditions. This would only lead to a large multiplication of results which is very undesirable.

It is therefore obvious that a general rule which shall be both simple and practical, and which will be perfectly accurate and perfectly applicable in every case, is an impossibility.

It must be recognised, therefore, that at the best all such results are in their very nature approximate only, since the deliveries depend on the interior diameter and the condition of the interior surface of the pipes, factors for which no possible theoretical allowance can be made otherwise than by practically testing every length of pipe before being used.

Even if this were done the results would be vitiated by alteration in the internal condition of the pipes after being fixed.

If, however, pipes of uniform quality are used throughout any installation, and if the rules given below are carefully applied throughout, the general balance of the whole system will be secured and the inevitable errors will not be sufficiently great to cause practical inconvenience, though the errors will prevent exact accuracy from being secured. Only in rare cases should any artificial adjustment of resistance be necessary, such, for instance, as the insertion of artificial checks, stops, or washers in a straight length of pipe. In many cases it may be necessary in complicated and highly sub-divided installations to apply valves of the double-regulation type to the radiators.

Indeed, it is open to question whether such a precaution is not advisable in every case in which accurate results are desired, in order to bridge the gap in size between two commercial sizes of pipe, to compensate for inevitable differences which will be found to exist in all pipes as generally manufactured, and generally to make it possible without trouble to regulate the flow through radiators in different parts of the system, without the expensive necessity of altering the pipe system once it has been put in.

It follows from what has been previously said that it is not essential for practical purposes that the difference in temperature between the inlet and outlet of a radiator should be absolutely the same as calculated.

If any one pipe in the system is slightly too small, the balance being otherwise good, the result is that the temperature difference in the radiators concerned will be greater than calculated. But the force of the circulation to those particular radiators is thereby increased.

On the other hand, if any particular pipe is too large, the excessive flow through the connected radiator will automatically increase the temperature of the return, and thereby diminish the circulating power through that circuit, though, as has been shown, it may also endanger the flow through other circuits.

In these respects, therefore, a system of pipes tends to be partly self-regulating in virtue of the natural properties of water.

These facts tend to minimise the results of inevitable inaccuracies, otherwise it is quite certain that a large proportion of the hot water installations put in would be total failures, for very few in this country, up to the present time, have been exactly calculated.

Unless the alteration in the temperature difference in a radiator is very great, the output of heat of the radiator is not diminished so seriously as to make the entire radiator a failure. The result of it may be a difference of one or two degrees in the temperature of the room. But the heat requirements of a room are in themselves so essentially uncertain that it is more or less a matter of chance whether such a difference as this is a serious matter or not. At any rate, the results of slight inaccuracies are not likely to be so serious as to make the entire radiator a failure, unless the effect of the mistake is increased by other mistakes in the calculation of heat losses (see Fig. 88).

Although the formula of Weisbach gives results which are incorrect for determining the absolute flow of water through pipes, yet when the values are systematically employed throughout an installation they certainly do secure a correct balance of the system of pipes. This is, of course, the essentially important point.

The values, therefore, of the resistances in straight lengths of pipe will be taken provisionally from Weisbach's formulæ until a complete set of results of experiments on the pipes as used in this country are available. The values can equally well be taken from any such results. Indeed, this method will enable any competent experimenter to design his own systems from the results of his own experiments, the nature of which will be explained later.

A demand by the engineer from the pipe makers for smoother tubes of more uniform bore and resistance would probably enable greater accuracy to be obtained than is possible with the pipes as now manufactured.

Resistances of Bends and Fittings.

It will be seen from the chapter on the flow of fluids that in the theoretical treatment of the subject, the values of single resistances are treated separately. The values of ζ there given represent the proportion of the total kinetic energy in one pound of water destroyed by a single resistance of that type.

This method in practice involves considerable trouble and complication, for every time the velocity in the pipe is changed different values are thereby assigned to the loss of head due to the single resistance.

In practical work, therefore, it is much more convenient to reduce these resistances to approximately equivalent length of pipe, so that the actual length of any section of pipe is slightly increased to allow for the single resistances, such as bends, elbows, etc., in that section.

The velocities of water in ordinary low pressure systems vary up to 6″ per second, and sometimes more. If the mean velocity is taken as 6″ per second, a simple calculation shows that the length of pipe which has approximately the same resistance as a single resistance, for which the value of ζ is equal to 1, is very closely in feet twice the internal diameter of the pipe in inches. Thus, at a velocity of 6″ per second the resistance of 6 feet of 3″ pipe is approximately equal to $\frac{v^2}{2g}$ or the velocity head of the water corresponding to the resistance of an obstruction where $\zeta = 1$. Similarly, 8 feet of 4″ pipe has the same resistance as one sharp elbow (see page 97 and Table XIII.).

Thus, to compensate for a round elbow it is sufficient to add to the length of pipe a number of feet equal to the diameter in inches.

It is not, of course, possible to ensure, without a great deal of trouble, that any apparatus will be constructed exactly according to the design, and there is therefore a certain amount of estimating necessary for the value of ζ for any section of a particular pipe. It is advisable, therefore, to be liberal, though not extravagant in the values allowed for ζ.

Resistances of Radiators and Boilers.

One of the elements of the total resistance is that of the radiators and boilers through which the water passes. This resistance in any case is small. Its actual value depends on the method of connection.

According to Recknagel, the resistance of a radiator or boiler is given in millimetres of water column by the following formula :—

$$p_e = 9.6\,(3\,n^3 - 3\,n^2 + 2\,n)\,v^2 = a\,v^2$$

where both connections are made at one side, and

$$p_n = 3.2\,(2\,n^3 - 3\,n^2 + n)\,v^2 = b\,v^2$$

where connections are made on opposite sides.

Where n is the number of sections.

v the velocity through the nipples in metres per second.

a and b having the following calculated values.

n =	2	3	4	5	6	7	8
a =	154	576	1,540	2,980	5,300	8,600	13,050
b =	19	96	269	576	1,055	1,750	2,690

n =	9	10	11	12	13	14
a =	18,820	26,100	35,000	45,700	58,600	73,600
b =	3,920	5,470	7,400	9,710	12,480	15,750

n =	15	16	17	18	19	20
a =	91,000	110,800	133,500	159,000	187,300	219,300
b =	19,500	23,800	28,750	31,100	40,400	47,500

The resistance of a radiator connected at opposite ends is thus given by Recknagel as smaller than the resistance of one connected at one end only, on the ground that the actual value depends on the passage through the nipples, and that when both connections are taken off at the same end, the water passing to the far end of the radiator has to pass through more nipples than if the connections are made at opposite ends.

This theory appears to be erroneous, for the circulation of water to the remote ends of a radiator is secured not by any circulating power external to the radiator, but by differences of temperature within the radiator itself, and the circulating power developed externally to the radiator will exhaust itself by short circuiting through the inner sections, when the connections are made at the one end.

With connections made at opposite ends, however, each path through the radiator has the same resistance.

Whether or not this view may be true, the resistance of a radiator is in fact so small that for practical purposes it may almost be neglected.

Recknagel gives the following as the actual value of the resistance in inches of water, when the radiator or boiler is supplied with sufficient circulation water to correspond to its surface with 36° difference between flow and return temperatures.

n =	5	10	15	20 sections
Total resistance (connected on one side)	.00164″	.0143″	.05″	.12″
Ditto (connected on opposite sides)	.000327″	.003″	.0107″	.026″

In the opinion of the present writer the resistance of a radiator can best be treated by allowing the value

$$\zeta = 2$$

for the entire radiator, and a similar figure will apply to the resistance of the boiler.

For the gross area of the sections of the radiator through which the water passes is so much greater than the area of the pipe, that it may safely be treated as a sudden enlargement of the pipe.

The general rule, therefore, for treating all such resistances is to add to the actual straight length of the pipe, a length corresponding to the single resistance.

This gives the total equivalent length of straight pipe, whose resistance corresponds to the actual resistance of the compound pipe consisting of straight pipe, bends, elbows, valves, and other resistances.

General Remarks on Methods of Determining Pipe Sizes.

It will be evident from the foregoing, that the actual principles on which the circulation of water depends are necessarily of considerable complication, and that while it is very desirable for practical reasons to obtain methods very much simpler than those which have been outlined, yet there is a limit to the possibility of simplification.

It is easy enough to determine simple methods of pipe sizing if one is content to disregard the question whether the sizes so determined are correct, theoretically or practically.

Such simplifications have, in the form of crude rules of thumb, been in use in this country since the method of heating by hot water was invented.

The essential questions in regard to the desirability of any proposed method of calculation are in their order of importance—

(1) Whether the system adopted will in all cases ensure that the apparatus will work so as to satisfy the purchaser, if not with theoretical accuracy? This is the first and essential condition.

It is far more important that the apparatus shall work to the satisfaction of the client than that the sizes of the pipes should be as small as possible, or that any theoretically desirable condition as regards temperatures or velocities shall be secured. The latter is merely a secondary consideration.

(2) It is, of course, important, especially in large jobs, that there shall be no waste of money in putting in pipes larger than necessary, but any saving in cost that might be effected in this way is very soon lost by a very small amount of alteration made to the installation after it is completed. The cost of such alterations is always high, and the necessity for them always entails discredit on the engineer who is responsible for the design.

(3) The matter may be also regarded in another aspect. The reduction in pipe sizes, in itself very desirable, might easily be secured at too great a cost in inconvenience, delay, and actual cost of making the calculation. To work out a theoretically accurate calculation of pipe sizes is necessarily a very tedious and prolonged operation. If the cost of working out such a calculation, besides causing some delay and inconvenience, is in excess of the saving in cost which that calculation will effect, it is a mistake to make it, provided that some simpler method will give satisfactory if not correct results. But if a simpler method cannot be certainly relied on to give satisfactory results, no reasonable amount of care in the preliminary calculations can be regarded as wasted.

The supreme object is to make the minimum amount of calculation necessary to secure that the apparatus shall work satisfactorily exactly as originally designed. The simplest form of calculation which will secure this result is the one which should be adopted.

At the same time it is very necessary that the expert engineer should have at his disposal the means for making the calculations as accurately as the present state of science permits, so as to enable him to discover the cause of any faulty working and to correct it when necessary.

Principles of Methods of Approximate Determination of Sizes of Pipes.

Before any progress can be made with the calculations for determining exactly the sizes of pipes, it is necessary to be able to arrive rapidly and without any considerable calculations, at approximate sizes for all pipes throughout the installation. This is necessary not only in order to enable the exact calculations to be completed by a process of trial and error, but also in order that it may be possible to prepare a scheme and estimate for any proposed work with as little labour as possible.

Any process of approximate calculation of the sizes to be ultimately adopted should involve an application of the principles already explained, but shorn of as much of their complication as possible.

It is easy enough to obtain an approximate method of pipe sizing which disregards those principles, by merely making the size of pipe proportional to the amount of heating surface it has to supply at the desired points, and disregarding both the circulating force available for conveying the heat to those points, and the amount of heat which will be lost on the way thither.

The use of any such method, however, without careful checking by accurate methods necessarily involves not only the danger that unnecessary expense may be incurred in providing pipes larger than required, but also that even though this expense is incurred the installation will not work satisfactorily. It is by no means true that the larger the pipes in a system, the greater will be the probability of a successful result. A successful result can be obtained much more certainly by correct proportion than by excessive size.

In any installation corresponding to any desired difference in temperature between flow and return there are four factors which ultimately determine the sizes to be adopted, and none of these can be safely disregarded. In the order of importance they are—

(1) The amount of heat required at the various points.
(2) The circulating power available for carrying the hot water there.
(3) The total length of pipe through which the water has to pass.
(4) The amount of heat lost on the way.

In selecting any system for approximating to the sizes, it is necessary clearly to bear in mind which of these factors are being disregarded in the system adopted.

A perfect system of approximation would take approximate account of them all. A less perfect system would disregard the last. Yet a less perfect system would disregard the last two. Only a very crude and imperfect system, which could hardly be dignified by the name of a system at all, would disregard the second as well. Yet such a system is and has been in common use in this country for many years, to the infinite detriment of the trade at large.

It will be the aim of the present discussion to explain systems of all these degrees of perfection or imperfection, and in the next chapter the method of correction will be explained, whereby the sizes may be determined as nearly as is practicably possible in the present state of knowledge.

General Observations on Systems of Pipe Sizing.

As explained above, the size of every main pipe should be such that it will be able to feed to *every radiator which it supplies*, the quantity of water necessary to keep it at the required temperature. The radiator which will generally be the most difficult to keep supplied with water is the "index" radiator, *i.e.*, that which is at the greatest distance from the boiler, measured along the pipe, and at the same time is the shortest distance vertically above the boiler.

If the pipes are so designed that this radiator does not get its full quantity of water, the sizing of the pipes will be in so far unsatisfactory.

It has been explained in the last chapter that the sizes of all pipes feeding this radiator must be determined with sole regard to its own position and without regard to the position of other radiators which may be supplied from the same pipes.

It is clear, therefore, that a correct system of determining the sizes involves that the position of this radiator only shall be considered in determining the sizes of *all* the pipes feeding it. This most unfavourably situated radiator A in any complete circulation will be called the "index" radiator of that circulation, and *it is the position of this radiator which determines the sizes of all the pipes feeding it, both flow and return.*

Barker's System.

Consider the general equation of a simple circulation, as shown on Fig. 122.

$$24\,h\,\frac{D_2 - D_1}{D_1 + D_2} = \rho\,\frac{l}{d}\,\frac{v^2}{2\,g}$$

This equation may be written

$$\frac{h}{l} = \frac{\rho}{24\,d}\,\frac{v^2}{2\,g}\,\frac{D_1 + D_2}{D_2 - D_1}$$

Apply this equation to two exactly similar installations, similar to Fig. 122, having one radiator each, the only difference between which is the height at which the radiator is placed. In all other respects the installations are the same.

If all artificial checks on the velocity of water in these two circuits are removed, and *if the same temperatures are maintained in flow and return,* the square of the velocity attained will be approximately proportional to the height h, since l and all the terms on the right side of the equation are the same in the two cases (except for a slight variation in the value of ρ which may be neglected). In other words, in these conditions the quantity of water circulated will vary as the square root of the height h. In practice an increase in velocity will cause a slight decrease in the difference of temperature, and therefore a decrease in circulating power.

Next compare two installations in all respects similar, except that the one in which the radiator is fixed higher has a proportionately longer run of pipes than the other, so that $\frac{h}{l}$ is the same in the two cases, all pipes being perfectly protected against loss of heat. The velocities maintained would be the same in the two cases. In this case the terms of the right hand side of the equation above are the same for the two installations.

The fraction $\frac{h}{l}$ is proportional to the available circulating power of the installation per foot run. In other words, each size of pipe has approximately the same capacity in B.T.U. for all cases where $\frac{h}{l}$ has the same value. The assumption is that the sizes of the pipes are so proportioned that the loss of head per foot run is approximately constant throughout the installation.

The criterion which determines whether one radiator is more unfavourably or favourably situated than another is therefore the value of the quotient when the height of the centre of the radiator above the centre of the boiler in feet is divided by the total travel of the water through the radiator in question.

The variation from theoretical accuracy in this theory is caused by difference in the loss of heat from the pipes. This will operate in two ways—

(1) By causing a small variation in the value of the fraction
$$\frac{D_1 + D_2}{D_2 - D_1}$$

(2) By increasing the water requirements of the longest circulation, and to some extent

(3) By causing a slight variation in the value of ρ.

The first two alterations are in opposite directions. With the overhead system the loss of heat in the pipes will favour the circulating power in the longest circulation, whereas the increased water requirements of that circulation will increase the friction.

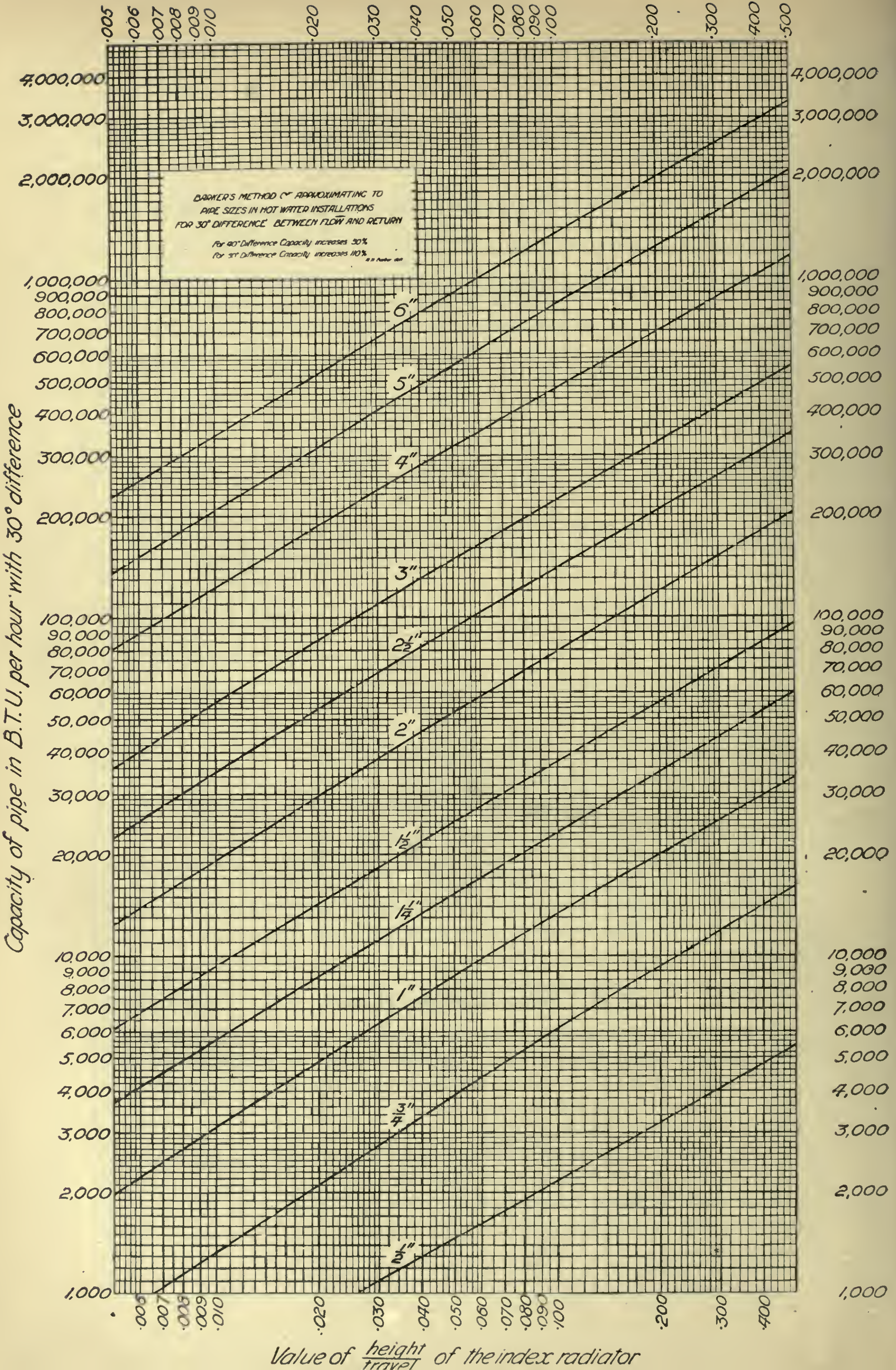
The assumption made in disregarding this difference is that these two contrary influences balance one another. At any rate, in determining which of two radiators is the index radiator of any particular circulation or installation, this question is not of sufficient importance to cause trouble.

It will thus be seen that for any given value of $\frac{h}{l}$ each size of pipe will have a definite capacity in B.T.U. which can be carried without generating a loss of head greater than the average throughout the circuit.

The table gives the *maximum* number of B.T.U. which may be carried by a pipe of standard size, so that the loss of head per foot run *may not exceed* the average figure.

The values are also plotted logarithmically on the diagram Fig. 127.

If this table is so used, that where a pipe has to carry a greater amount of heat than the figure corresponding to any particular pipe size, the next larger size is chosen, there will clearly be a surplus of circulating power


BARKER'S METHOD OF APPROXIMATING TO
PIPE SIZES IN HOT WATER INSTALLATIONS
FOR 30° DIFFERENCE BETWEEN FLOW AND RETURN
For 40° Difference Capacity increases 30%
For 50° Difference Capacity increases 110%
Capacity of pipe in B.T.U. per hour with 30° difference
Value of height/travel of the index radiator
6″
5″
4″
3″
2½″
2″
1½″
1¼″
1″
¾″
½″
.005
.006
.007
.008
.009
.010
.020
.030
.040
.050
.060
.070
.080
.090
.100
.200
.300
.400
.500
4,000,000
3,000,000
2,000,000
1,000,000
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700,000
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500,000
400,000
300,000
200,000
100,000
90,000
80,000
70,000
60,000
50,000
40,000
30,000
20,000
10,000
9,000
8,000
7,000
6,000
5,000
4,000
3,000
2,000
1,000

throughout the circulation, owing to the fact that all the pipes will be carrying amounts of heat less than they would be able to carry under the given conditions.

The use of such a table, therefore, would give sizes on the whole slightly larger than necessary, but in no case would any pipe be more than one size too large.

It would be possible to use such a table in a yet more exact manner, by dividing any given pipe length into two parts, of different diameters, such that the mean resistance of the whole pipe length would be equal to the average loss of head per foot run throughout the whole circulation.

This, however, would probably involve too much labour for an approximate determination. If the table is used in such a way that no pipe in the index circulation is to carry more than the given quantity of B.T.U. the results obtained will generally be moderately correct for the index circulation.

Parts of the index circulation will be common to both that circulation and others having greater circulating power.

If the same table is used without alteration for determining the range of sizes to be employed for that part of the second circulation which is not common to the index circulation as well, the sizes so determined will be too large, and may often be considerably too large.

Take for example the installation shown on Fig. 118. Here the sections composing the index circulation are as follows :—

(*a*) BF, FP, PP_1, P_1B.

The parts composing circulation No. 2 are as follows :—

(*b*) BF, FR, RC, CB.

The following parts are common to both these circulations :—

(*c*) BF, FP, P_1B,

and only the following parts belong exclusively to circulation 2 :—

(*d*) PR, RC, CP_1.

This inaccuracy of principle may be allowed for by calculating the value of l_2 for the second radiator R from the formula

$$l_2 = (l_1 - \lambda_1)\frac{h_1}{h_2} + \lambda_2$$

where λ_1 λ_2 are the lengths of the respective circuit pipes, PP_1 and $PRCP_1$ and l_1 l_2 are the equivalent travels.

Now for approximate calculations only, the sizes of parts mentioned in (*a*) above are determined from the value of $\frac{h_1}{l_1}$ for the index radiator. The parts (*d*) are determined from the value $\frac{h_2}{l_2}$ for radiator No. 2 where l_2 has the value obtained from the formula above. The resistance of parts (*a*) is smaller than it would be if calculated throughout from the values of $\frac{h_2}{l_2}$ for radiator No. 2. Since the range of pipe sizes is only a mean for the entire circulation, it will be evident that this approximation, when used to calculate the sizes of parts (*d*) would give values larger than they need be, and therefore are well on the safe side for estimating purposes. If the installation were carried out according to this approximation and without reducing the value of l_2, circulation No. 2 would be in danger of short circuiting the index radiator. It is essentially for this reason that the sizes given by this or any other approximate method require to be corrected by accurate calculations, such as will be explained in the next chapter.

Loss of Heat from Pipes.

This loss can be approximately compensated for by adding to the quantity of heat estimated to be required at the given points, a certain proportion of the length of the travel for each radiator. If this is done the effect of it is that the pipe sizes for each circulation are in fact calculated not only according to the number of B.T.U. required at each point, but according to a rough estimate of the amount of water which will have to pass through the pipes. This calculation as regards the main pipes will of course only have to be made for the index radiator.

The heat required from all other circuit pipes, other than that belonging to the index radiator, will only be increased according to the length of the circuit pipe. The factor by which the length of travel or the length of the circuit pipe, as the case may be, is to be multiplied, represents a guess at the average number of B.T.U. which will be lost from every foot of the pipe comprising the circulation.

The figure will be different, therefore, according as the pipe is intended to be coated or not. For a very long circulation the figure will be greater than for a short one, because the average diameter of the pipes in a long circulation will generally be greater than for a shorter one.

The following figures are recommended as average values :—

For an uncoated circulation up to 100 feet long, 100.
For an uncoated circulation up to 200 feet long, 120.
For an uncoated circulation up to 500 feet long, 160.

For coated pipes half these figures are to be used.

The process, therefore, of using this table is as follows :—

(1) For every radiator in the entire system calculate the value of $\frac{h}{l}$.

For every pipe in the entire system calculate the number of B.T.U. it carries by addition, after increasing the B.T.U. carried by the index circulation by the length of travel multiplied by the given factor, and the B.T.U. carried by all other circuits by the length of that circuit multiplied by the given factor.

The diagram will then give the sizes of every pipe in the system, which sizes will be well on the right side.

The capacities given by the diagram may be increased by 50% for 40° difference and by 110% for 50° difference.

The above method of approximation is probably as accurate as it is possible to get by an approximation, without going fully into the circulations. This process is certainly more complicated than is desirable, but it must be clearly pointed out that any simpler process involves the sacrifice of accuracy of principle to a greater degree.

Rietschel's System.

The next system of approximation which will be explained will determine the size of the risers according to the height only, the length of travel being disregarded except in so far that different capacities of pipe sizes are given for the same height of radiator, according as the riser is situated at a greater or less distance from the boiler. This method is due to Rietschel, and is probably that which combines to the greatest degree ease of application with moderate accuracy of principle.

A liberal and fairly intelligent use of this system, however, may be relied on to give approximately correct values, although this system also requires to be subsequently corrected by making exact calculations.

According to the tables, it is only necessary here to measure the total length of the installation and the height of the radiator. Table XXX. gives directly the number of B.T.U. corresponding to these figures for each size of pipe above 1″.

The chief drawback to this system is that it involves a multiplication of tables, which is not in itself desirable.

Method 3.

This is still a simpler method, and is due in the first place to Birlo. Birlo gives the capacity of any size of pipe feeding a radiator whose height is h above the centre of the boiler as follows, for risers only :—

$$d'' = c\sqrt{\frac{W}{h^{0.4}}}$$

where $c = 0.0163$ for underground mains and $c = 0.0151$ for overhead mains.

The sizes of the horizontal mains are obtained by adding the corresponding areas of the risers. For risers at the remote ends of large installations, the next size larger is taken as the required pipe size.

This method is more in the nature of a rule of thumb, but it is largely used with satisfactory results on the Continent.

Table XXXI. gives the capacities of various sizes of pipe on this system.

CHAPTER XXV.

THE PRACTICAL METHOD OF MAKING EXACT CALCULATIONS OF PIPE SIZES.

In the previous two chapters have been explained the principles according to which the sizes of pipes in a hot water installation are determined, and the principles of the approximations adopted. In the present chapter will be discussed the practical methods of applying these principles as accurately as is practically possible.

The carrying out of the necessary calculations is sometimes somewhat confusing and tedious. Unless the whole of the installation is shown in its correct proportions on *one* drawing, a great waste of time is involved in referring backwards and forwards from one plan to another. This may be avoided by providing an "isometric projection" showing the entire plant to scale exactly as it is proposed to construct it. On this drawing the result of every calculation as it is made can be marked.

Those who are unfamiliar with this invaluable system of projection should study some good work on geometrical drawing. The method is easy to acquire, and every heating engineer should be thoroughly familiar with it.

On this isometric projection the number of B.T.U. required at each point should first be marked in red ink, the height of the centre of each radiator above the centre of the boiler in feet in black ink. The drawing should also show which of the pipes are to be coated and the assumed temperature of the air surrounding each pipe.

The "travel" of each pipe is also measured and marked, that is, the *greatest* distance which *any of* the water passing through that pipe has to go from the boiler back to the boiler.

The next process consists of an approximate determination of the sizes of the pipes throughout the entire installation by one or other of the methods described in the previous chapter. The simplest of these methods, namely, No. 3, is generally sufficient for ordinary small installations, but where an installation is widely extended or has unusual features, method No. 1 should be employed.

This determination will generally have been already made for the provisional scheme required for tendering purposes, and in order to complete the exact calculation, the sizes so determined will usually suffice.

These approximate sizes should be marked on the projection in pencil, so that they can easily be altered when necessary.

The approximate amount of heat lost from each length of pipe in the entire installation must then be determined.

Numbering of Pipes.

For this purpose, and for the purposes of subsequent calculation, every undivided length of pipe must have assigned to it on the projection a definite number. Each section of each pipe should be numbered in a systematic manner, in such a way that the pipe between every pair of junctions is represented by a separate number. This number refers only to the length of pipe between these two junctions. A beginning may be made at the main boiler flow, the numbers increasing as the distance from the boiler increases, or any other systematic plan may be adopted.

In thus labelling the pipes it is not necessary to give a pipe two numbers because it is divided by a bend or elbow. Any pipe, however irregular, through which the same water passes should have only one number if it is of uniform diameter. Each connection to each radiator should have its own number, and every radiator should also be numbered.

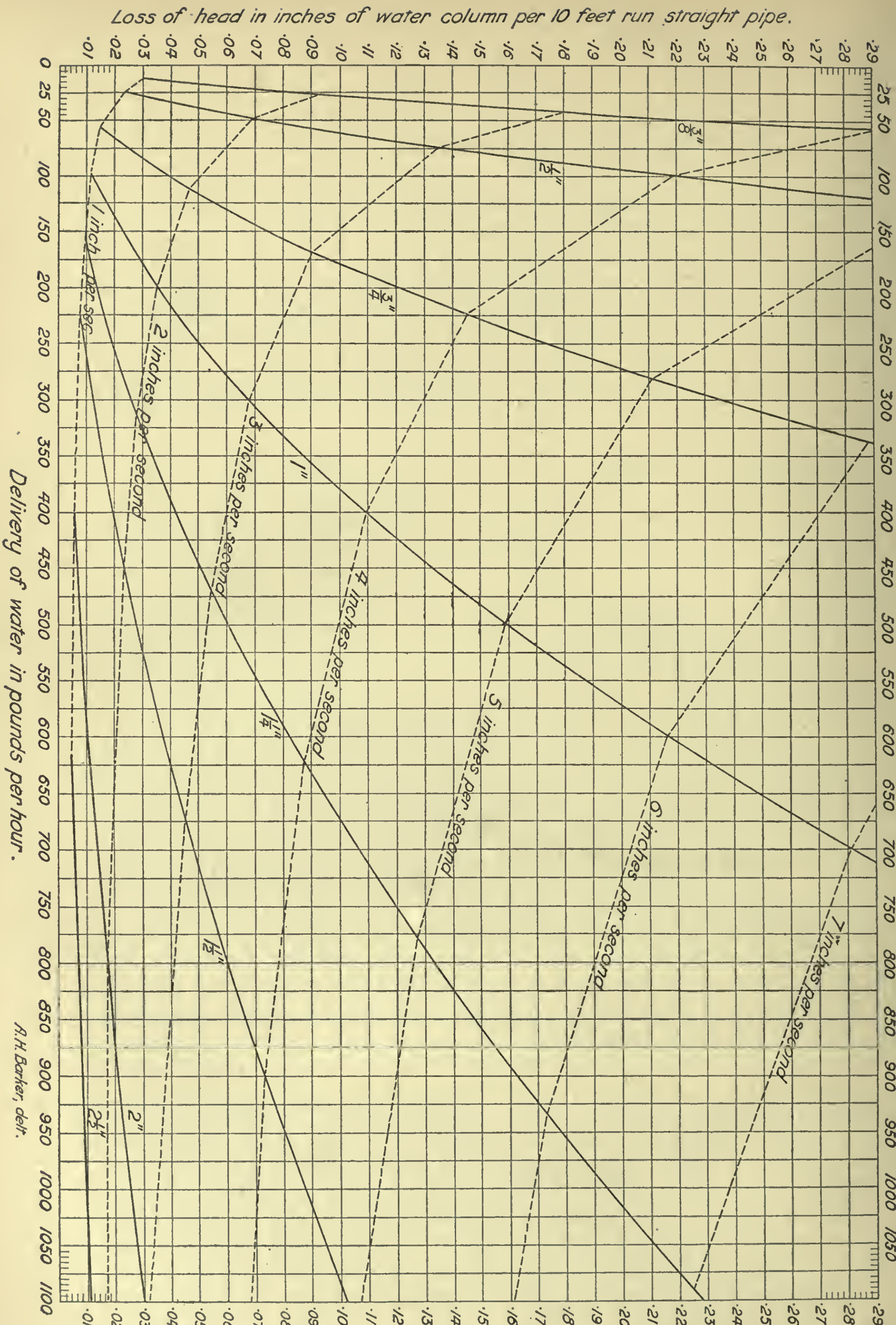
The length of and loss of heat from each section of pipe represented by a separate number must then be separately measured and calculated, which can be rapidly done by means of the tables. These results are also marked on the plan in black and red respectively.

In making this calculation it may be assumed that the temperature of the water in all flow pipes is equal to the boiler temperature, also, the temperature of the water in all return pipes may be calculated as being equal to the return temperature.

Estimation of Total Heat Carried by each Pipe.

These figures may now be obtained by starting at the most remote radiator and adding up in succession the quantities of heat lost from each radiator and numbered pipe length. It will be first necessary to calculate the heat lost from each *circuit* (of course, including the radiators) before making any additions for the mains.

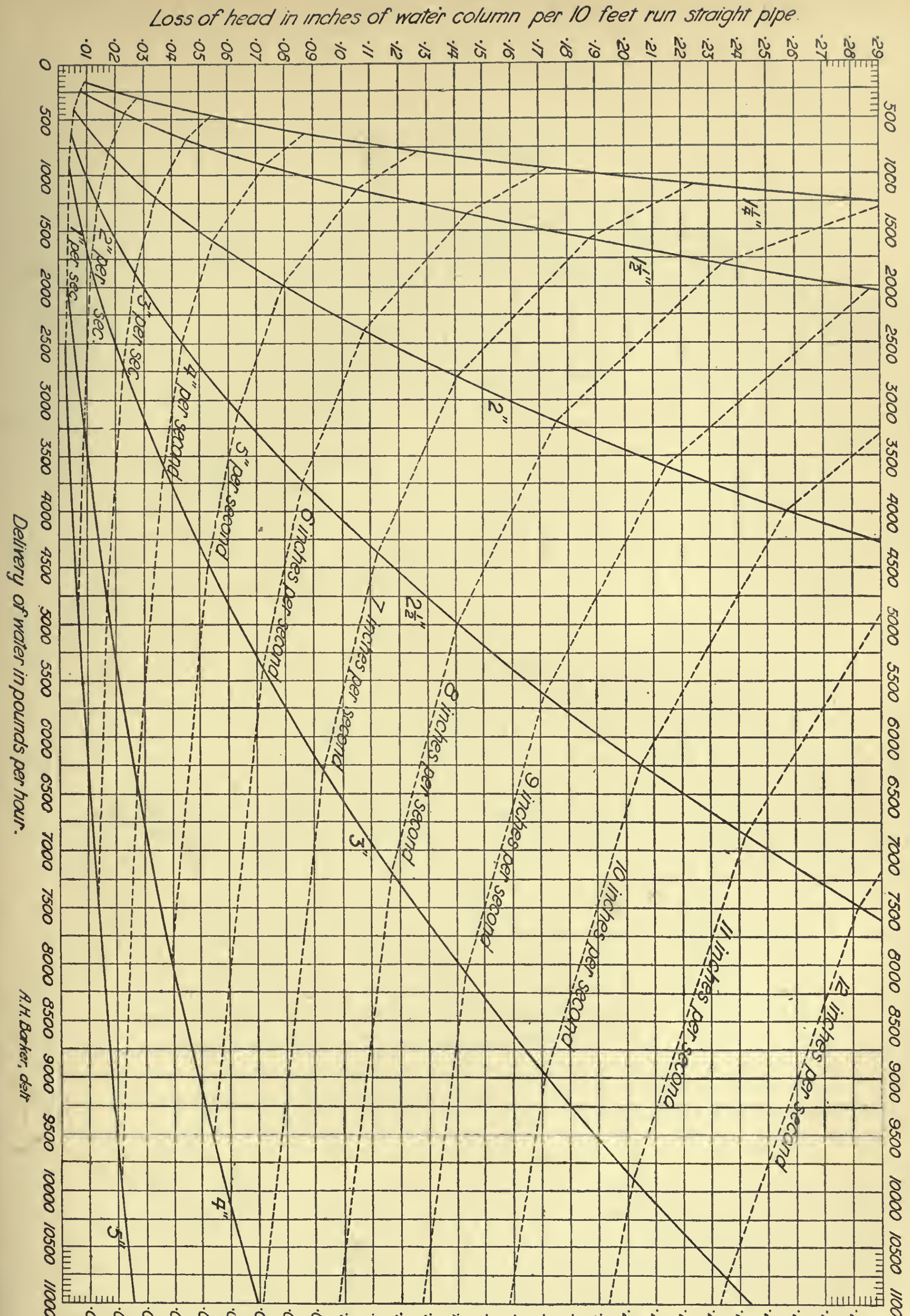
Where two circuit pipes combine to form a main, their two respective quantities of heat should be added together, and the result figured in red



FRICTION OF WATER IN PIPES ACCORDING TO WEISBACH

1. Low velocities 1" to 7" per sec., small pipes, $\frac{3}{8}$" to $2\frac{1}{2}$" dia

FIG. 128.



FRICTION OF WATER IN PIPES ACCORDING TO WEISBACH

II. Low velocities, 1" to 12" per sec., large pipes 1¼" to 5" dia.

FIG. 129.

pencil on the end of that portion of main remote from the boiler. To the figure so obtained should then be added the quantity of heat lost from that length of pipe, and the result figured on the plan at the end of that pipe length nearest the boiler.

The total amounts of heat carried by each pipe leaving the main at that junction should then be added to this figure, and the result figured on the remote end of the next pipe length, and so on back to the boiler. By this means the amount of heat carried by each pipe is obtained with some accuracy.

It may be necessary to make some small alteration in these figures at a later stage, but this is not generally necessary unless great accuracy is looked for.

All these figures are preferably marked on the plan in red pencil, so that they can be removed and corrected without trouble if required. When any quantity of heat is definitely determined it should be figured on the plan in red ink.

Quantity of Heat Carried by each Pipe in the System.

It is next necessary to determine the number of pounds of water desired to flow through each pipe in the entire installation.

In order to do this it is necessary to work the opposite way round, that is, to start at the boiler along the main flow pipe.

The red figure on the main flow and return pipes near the boiler gives the approximate total loss of heat from the entire system, including pipes, both flow and return. The desired difference of temperature between the flow main and the return main, connected to the boiler is known. A simple calculation then gives the quantity of water which must pass *through the boiler* per hour, from the formula

$$H = Q\,(t_f - t_r)$$

This should be figured on the plan in green pencil.

The quantity of water passing through the boiler will, of course, be undiminished as far as the first branch from the flow main, where it will divide into two or more portions. This total quantity of water must then be divided between these two or more branches in the same proportion as the total B.T.U. carried by each.

The formulæ for making this division is $q_1 = Q \dfrac{W_1}{W_1 + W_2}$ and $q_2 = Q \dfrac{W_2}{W_1 + W_2}$ where Q = total lbs. of water, q_1 = lbs. of water through section 1, W_1 = total heat carried by section 1, etc. Then taking each branch in turn, similar calculations are made at each junction for the whole of the flow pipes in the entire installation, and the figures marked on plan in green pencil. The figures so marked on the plan will give fairly closely the actual quantity of water required in each branch.

The quantities for the return branches can be obtained by merely adding the corresponding figures already determined for the flow branches. The quantity of water is not, of course, reduced by passing through a radiator or through any system of pipes.

This, therefore, gives the quantities of water passing per hour through every pipe of the whole installation.

Determination of Temperatures in Pipes.

If an accurate determination is required it is next necessary to calculate from these figures the temperature at each point, *i.e.*, to correct the original guess at the same figures. These will be calculated for each pipe length, again starting from the boiler flow, from the formula

$$Q\,(t_1 - t_2) = K\,c\,l \left(\frac{t_1 + t_2}{2} - t_a \right) \text{ or approximately } \frac{K\,c\,l\,(t_1 - t_a)}{Q} = t_1 - t_2$$

Assume, for instance, that the main flow pipe temperature is 180°, that the first unbranched length of flow pipe is 36′ long, that its diameter is 3″, that the flow in pounds per hour through it is 6,460 lbs., the surrounding temperature being 50° F., and that it is coated with a material having an economy of 70%, that is to say, that the coated pipe will only lose 30% of the heat that would be lost by the same pipe uncoated.

We have then the total loss of temperature in this pipe approximately

$$\frac{2.05 \times 33 \times 130}{6{,}460} = 1.37^\circ$$

showing that the temperature at the first junction is 178.63°

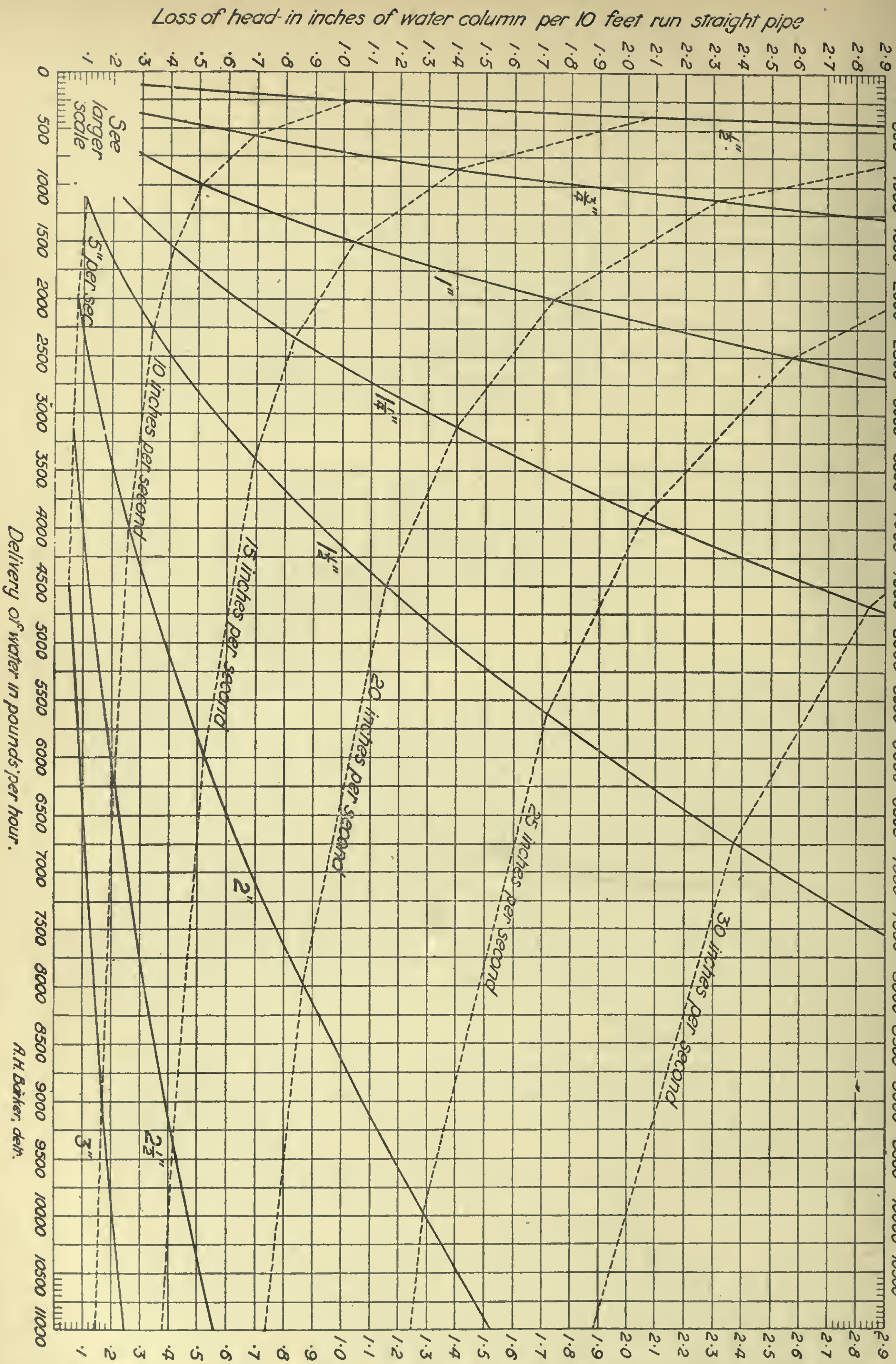
These temperatures are figured on the plan in blue pencil.

The same process should be carried out throughout the entire installation as far as the junction into the return main.

If the calculations have been correct these results will show that all return pipes have approximately the same temperature.

If it is found that the temperature of any return branch is too high or too low this points to a mistake in the calculations, and the quantities of water calculated for the various branches are to be readjusted so that all the return temperatures are approximately the same within one or two degrees.

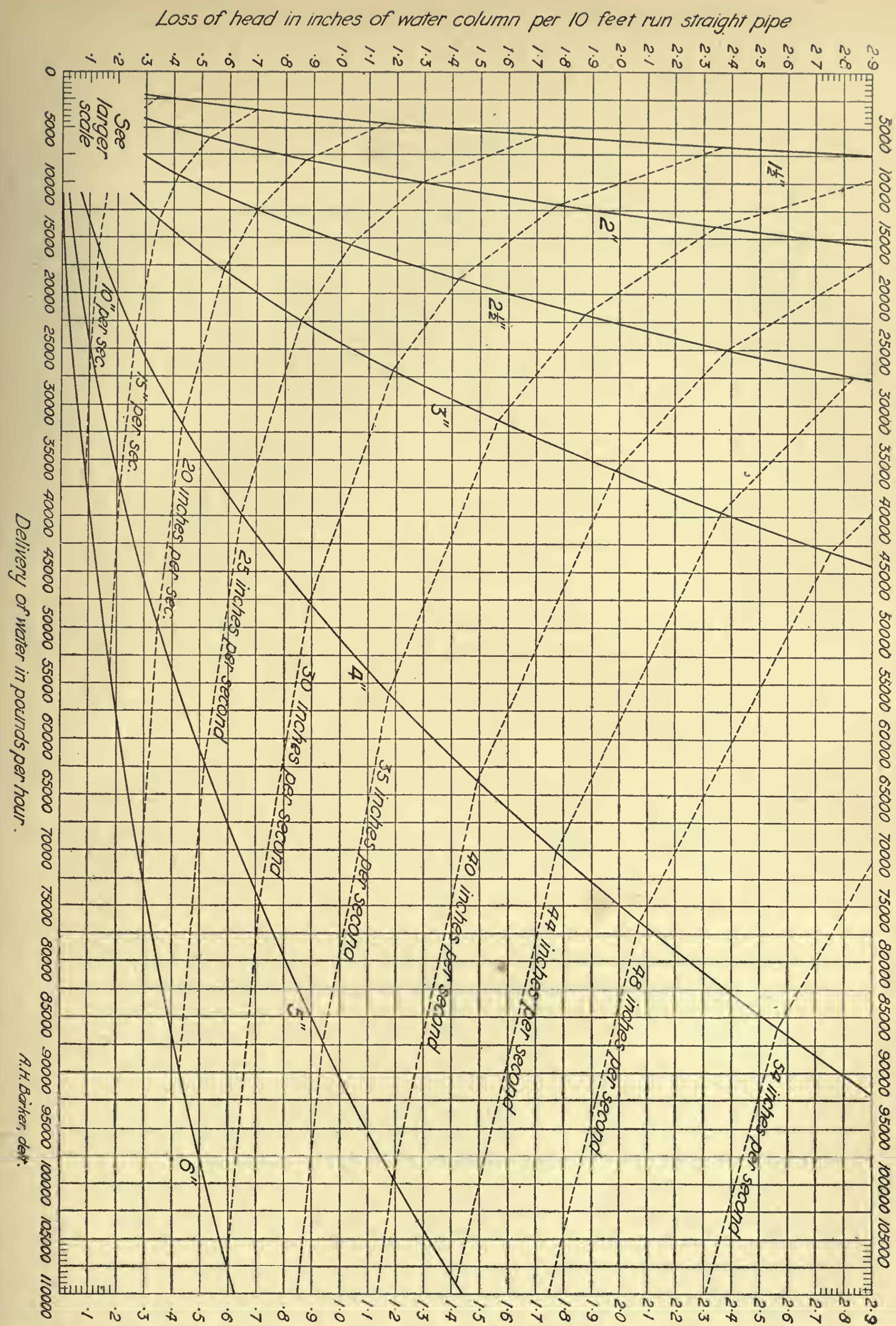
Any alterations in pipe sizes which are subsequently made on account of the friction in any branch being too high or too low, will not generally so far affect the quantities of heat or water as to render necessary any alteration being made in these quantities, though if great accuracy is desired the calculations should be readjusted throughout.



FRICTION OF WATER IN PIPES ACCORDING TO WEISBACH

III. Medium velocities. 5" to 30" per sec., small pipes, ½" to 3" dia.

FIG. 130.



FRICTION OF WATER IN PIPES ACCORDING TO WEISBACH

IV. Medium velocities. 10" to 54" per sec., large pipes, 1½" to 6" dia.

FIG. 131.

It is not generally necessary to calculate the exact temperatures of the return mains, provided that the temperatures of the outlets from all radiators are approximately the same.

So far as the calculations have now proceeded, they have been made with reference only to the quantities of heat *desired* from the various radiators, and the corresponding quantities of water *desired* in the various pipes, and the results of the calculations of measurements have been figured on the isometric projection.

Calculation of the Loss of Head by Friction.

The next step is to calculate the amount of loss of head by friction when these desired quantities of water pass through pipes of these sizes and to compare this loss of head with the available circulating power. These calculations are made on a specially ruled form of which the ruling is shown below.

If these calculations show that the pipe sizes provisionally chosen are either unnecessarily large or too small, suitable corrections must be made in order to increase or decrease the total resistance of the various circulations as may be necessary to bring the friction in the pipes to the same value as the available head.

This calculation must be made in a systematic way, so that all parts of the installation are brought under review. The method according to which this is done will now be explained.

It has been pointed out that in any circulation the circuit pipe which will be most difficult to keep supplied with water will be that connected to the "index" radiator, viz., that for which the ratio of the height above the boiler to the travel has the lowest value.

It is the main object of the design to secure the result that all pipes in the entire installation shall be suitably fed. From this and from the explanations in the previous chapter it follows that all the pipes feeding the index radiator must be calculated with regard to the circulating power attainable from the index radiator. If this is not done, it is quite certain that the index radiator, at any rate, will not receive its proper quantity of heat. Thus, circuits which join the index circulation at a higher level, and which have greater circulating power than the index circuit, must be calculated out according to the circulating force available in those circulations.

Parts of these circulations are, of course, common to both them and the index circulation, and the sizes of these common parts have already been determined according to the circulating force derived from the index circulation.

A separate calculation must be made for every circulation into which the entire installation can be divided. All these calculations will be made on separate copies of the form, of which a specimen ruling is shown below.

RULING OF FORM FOR EXACT CALCULATION OF PIPES.

Name of Building :— Architect :—													Trial Alterations.						
1	2	3	4	5	6	7	8	9	10	11	12	13	4a	5a	7a	8a	9a	10a	13a
Number of Pipe.	B.T.U. Carried.	Quantity of Water.	Proposed Diameter.	Approximate Velocity.	Length of Pipe.	Value of Single Resistances. $\Sigma\zeta$	Length in Straight Pipe Equivalent to Single Resistances.	Total Equivalent Length. 6 + 8.	Resistance per 10ft. Length.	Total Resistance. 10 × 9 × 0.1.	Available Head. $24h\frac{D_2 - D_1}{D_1 + D_2}$	Required Alterations Proposed. 12 − 11.	Altered Diameter.	Approximate Velocity.	Single Resistances.	Equivalent Length in Straight Pipe.	Total Equivalent Length.	Resistance per 10ft. Length.	Nett Alterations.

Calculations for Index Circulation.

The first process in making the calculations for any complete circulation is to obtain the circulating pressure forcing the water through it. The general method by which this is done in practice has been explained in the previous chapter, namely, for a two-pipe system to assume that the water retains the boiler flow temperature as far as the centre of the radiator and on the return side at once assumes the temperature of the boiler return.

With these assumptions the circulating pressure in inches of water is given by the formula

$$24\,h\,\frac{D_2 - D_1}{D_2 + D_1}$$

These values can be at once obtained from the table, by multiplying the corresponding figure by the height between the centre of the boiler and the centre of the radiator. This figure should be written in column 12 at the top of the form representing that particular circulation.

For a single pipe system it is generally sufficient to calculate the height of the centre of gravity of the connected radiators and apply the same formula.

It is now necessary to assume that the required quantities of water are flowing through each portion of the pipe composing that circulation, and to calculate the amount of head which would be lost by friction in each numbered pipe length if the sizes are as originally figured.

These figures are obtained by multiplying the figure obtained from Table X. or from Figs. 128-131 by the length of the pipe, divided by 10. The figures so calculated are written in column 11 of the form.

All the lengths of pipe comprising the complete index circulation should first be figured in their proper order in the first column, and the particulars for filling up the remainder of the columns can be taken off the isometric projection.

To the actual length of each pipe is added a certain calculated length whose resistance would be equal to that of the various fittings. This is done by means of the table given on page 97 and Table XIII. After this is done the total loss of head per 10 foot run of each pipe length can be obtained from the diagram, and the actual loss of head calculated.

A trial addition is then made of all these losses of head throughout the whole of the index circulation. If the sum is found to vary considerably from the available circulating head which has been written at the head of the calculations, corresponding alterations are to be made in one or more of the lengths of the pipe, so that the total calculated loss of head is approximately equivalent to the available head.

It is evident that it will be very advisable for the sake of economy to increase the size of the circuit pipe feeding the index radiator, so that there is very little pressure lost in that individual pipe. This will allow of a greater loss being distributed throughout the remainder of the circulation and so favour economy. In deciding which length of pipe to reduce in size if the calculations show a surplus of available head, regard should be had to the question of cost. That length of pipe should be reduced which will save the greatest cost.

The right hand half of the sheet is left for the trial corrections, and the adjustments should be made until there is very little difference between the available head and the head lost by friction.

If this calculation results in considerable alteration being made in the pipe sizes, it may be advisable to readjust either the calculated quantities of heat lost from the pipes or the quantities of water, and to make the whole calculation again. But this is not generally necessary in practice, except in very large or important installations.

When the calculations for the index circulation are completed the sizes of the pipes comprising that circulation may be regarded as fixed, and the next circulation proceeding inwards towards the boiler is then calculated. The first process in each case is to calculate the pressure circulating head of that circulation by means of the formula

$$24\,h\,\frac{D_2 - D_1}{D_2 + D_1}$$

and this value is as before figured at the head of the sheet of calculations referring to that circulation in column 12.

Each pipe composing circulation No. 2 is then figured on the form in exactly the same way as has been already done for the index circulation. The calculations have been already made for most of these sections which are common to circulation No. 2 and the index circulation, and they can be immediately written in.

The only part of circulation No. 2 which has not been calculated is the circuit pipe. The resistance of the circuit pipe must be increased, that is to say, the size of the pipe must be reduced, so that the total resistance of the entire circulation No. 2, including the part already calculated, is equal to the available head for circulation No. 2. This will often result in a surprisingly small pipe being required for the circuit pipe.

If the calculation gives a size which is smaller than it is thought advisable to put in, owing to practical considerations, provision should be made of a valve having double adjustment on the radiator, so that the resistance can be increased artificially if necessary.

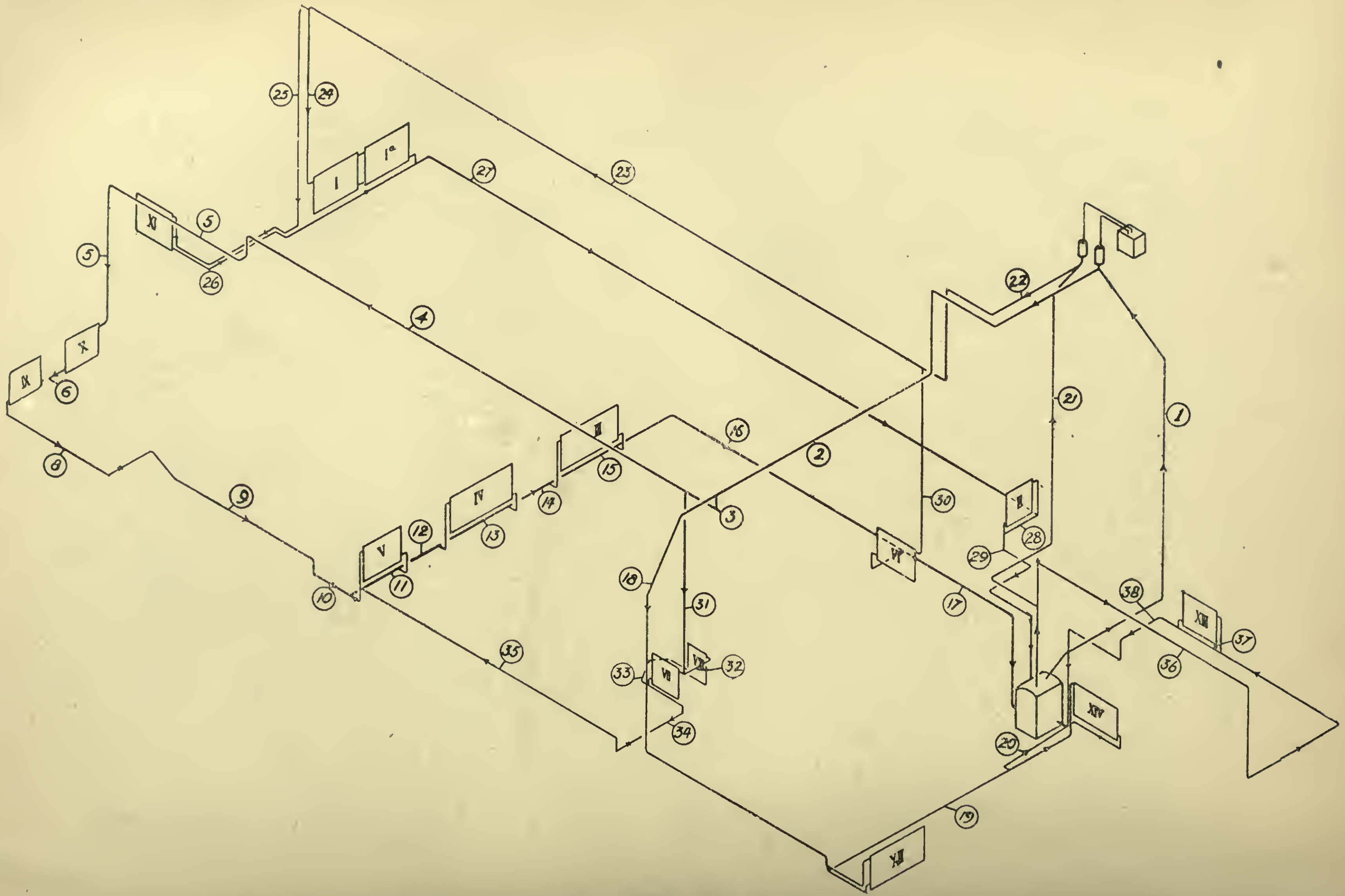
The same process is carried out for every circulation comprised in the entire installation.

The calculation may be regarded as complete when there is no greater difference between the available head, and the head practically calculated than 5%. It is a waste of time to make the calculations agree exactly, since the exact value of the friction cannot be determined with mathematical accuracy.

In order that the above remarks may be understood, an outline is given of the calculations necessary for an actual installation having unusual or peculiar features. If this is understood no difficulty will be experienced in dealing with a normal installation.

Calculation for an Irregular Installation.

In many or most cases the installation which has to be figured for pipe sizes, is of a special or irregular type, and the heating engineer cannot be quite sure how far the ordinary rules of thumb for the determination of pipe sizes are applicable to special cases, such as are of frequent occurrence in practice.



I
Ia
XI
X
IX
III
IV
V
II
VI
VIII
VII
XIII
XIV
XII

FIG. 132.

It has therefore been thought advisable to take the case of a small actual installation of irregular type and explain in detail the method by which the sizes are arrived at and checked, and to remark on the practical considerations which have frequently to be taken into account.

The installation is that in a small church, which had to be carried out on the most economical lines with a minimum amount of trenching, and with all radiators and pipes as inconspicuous as possible.

As is frequently the case in a church or other artistic building, there was great difficulty in finding space in which to place the radiators, and the sizes of the radiator had to be determined according to the space available. The internal temperature required was 55° with an outside temperature of 30°. The application of the ordinary rules gave the following as the heat requirements :—

Church nave	18,500 B.T.U.
Chancel and aisles	74,600 B.T.U.
Clergy vestry	9,380 B.T.U.
Choir vestry	7,700 B.T.U.
Ventilation	116,000 B.T.U.
	226,180

The general arrangement of the pipes and radiators is shown in Fig. 132.

The largest size of radiators which could be fixed in the various positions were necessary to get the bare heating requirements, even at a moderately high temperature in the radiators.

The only radiators that were sanctioned by the authorities in the nave itself were Nos. 3, 4 and 5, and as these were fixed at the back of pews they were not allowed to be raised to a high temperature, for fear of causing discomfort to the persons sitting immediately in front of them.

Special sanction was obtained from the authorities for carrying pipe 2 across the church above the arch separating the nave and the chancel, the specified condition on which the pipe was sanctioned being that it should not exceed 2″ in diameter.

Sanction was also given to carrying pipes 4 and 23 in the space between the rafters over the aisle. Special sanction was also required for fixing one pipe above the floor next to the seats at 9, 35 and 27, with the condition that these pipes were not to be made very hot.

Great difficulty was experienced in finding room for the expansion tank.

These details have been given in order to illustrate the conditions and limitations under which the heating engineer has to work, and according to which he has to calculate his pipes.

The following explains the process of determining the various unknown quantities :—

(I) Selection is made of the maximum temperature at the boiler of 180° F., and the maximum difference of temperature 50°. The total heat requirements of the building being 226,000 B.T.U. and all the pipes being exposed within the building itself.

The total amount of water required to be circulated is $\frac{226,000}{50} = 4,520$ pounds per hour through the boiler.

The maximum surface which could be got in under the limitations imposed is shown below.

The whole of the pipes were exposed and were intended to be used as heating surface, and as they provided a considerable portion of the entire quantity of heat required, it is obvious that they cannot be neglected.

It will be seen here that the B.T.U. required at each point are not given, but the greatest possible number of square feet in each radiator which could be accommodated in the various positions was determined by the local conditions. The type of radiator was determined by the architect. These were National Radiator Company's "Tuscan Flue Type."

In order to commence the calculations it was necessary to make a preliminary guess at the mean temperature which could be maintained in the radiators. From this the number of B.T.U. emitted by each was estimated, so as to obtain some idea of the relative emission of heat from each circulation. These were provisionally determined, as shown in the table.

Rad. No.	Sq. ft.	First estimated difference of temp.	Value of K.	B.T.U.
I. and I.*a* ...	240	90	1.3	28.000
II.	120	70	1.2	10,100
III.	90	70	1.2	7,600
IV.	90	70	1.2	7,600
V.	48	75	1.22	4,400
VI.	48	100	1.33	6,400
VII.	36	95	1.31	4,500
VIII.	36	95	1.31	4,500
IX.	48	85	1.27	5,200
X.	72	95	1.31	9,000
XI.	72	75	1.22	6,600
XII.	60	90	1.30	7,000
XIII.	48	90	1.30	5,600
XIV.	60	70	1.2	5,000

Circulation No. 2 carries, in all, 54,800 B.T.U. in radiators, and circulation No. 22 carries 53,500. Hence, assume the total quantity of water is divided about equally between the two branches.

It was also necessary to make a provisional determination of the sizes of all the pipes in the system. A guess was first made at the sizes of pipes having regard to the probable approximate number of pounds of water per hour which should be circulated through them on the above assumption. The velocity assumed was 5″ per second.

This gives a provisional diameter for all pipes in the system. Pipe No. 2 was limited by the architect to 2″, also pipes 4 and 23. The remainder of the pipes were provisionally guessed at on the above basis.

Next the surface of all the pipes so arrived at was calculated. A guess was made at the temperature of the water at each junction of the pipe. Then the total provisional heat lost from each section was calculated.

It will be seen that the entire pipe installation for the body of the church is divided into two parts, the system beginning sections 1 and 2, ending at 17, and a second beginning 21 and 22, and ending at 29.

On this estimated basis the total B.T.U. carried by the first of these pipes and radiators at their estimated temperature, was found to be 137,000 B.T.U., and the heat lost from the other section was found to be 122,000 B.T.U. It is evident, therefore, that the number of B.T.U. required can be obtained under the given conditions.

The amount of water required in this instance to flow up along pipes 1 and 2, as far as the junction with 3, is about 2,200 pounds per hour.

The circuit comprising 18, 19 and 20, and the two radiators Nos. 12 and 14, will carry approximately 45,000 B.T.U., while the other circuit, consisting of pipes 3, 4, 5, 6, 7, etc., will carry approximately 92,000 B.T.U.

The total quantity of water, 2,213 pounds, should therefore be divided between these two circuits as follows :—

$$2{,}200 \times \frac{92{,}000}{137{,}000} = 1{,}480$$

Circuit 3, 4, 5, carries 1,480 pounds, and circuit 18, 19, 20, 720 pounds.

These quantities of water may now be regarded as finally determined, seeing that the system is not of any considerable complexity.

It is now necessary to determine the actual loss of heat, given this flow of water through the pipes. This can be exactly determined by the formula

$$t_r = \frac{2\,Q\,t_f - S\,K\,(t_f - 2\,t_a)}{2\,Q + S\,K}$$

From this formula the temperature at this point 2, 3, is found to be 171° when the temperature of the water leaving the boiler is 180°. Similarly, the temperature at junction 5, 4 is found to be 166°, and at the junction of the radiator 163°.

After passing the radiator the temperature is calculated at 152° in the following way:—

Assuming that 1,380 pounds of water flows past the radiators, both they and the pipes having an average temperature of 155°, which will be approximately their actual temperature, they will give off 15,200 B.T.U. The temperature, therefore, is found as follows:—

$$163 - \frac{15{,}200}{1{,}380} = 152$$

By carrying this process further the temperature at point 10, 11, is found to be 149°.

A similar process carried round the circuit 31—35 gives a temperature of 150° at the same point, which is approximately correct, and does not call for any alteration.

A similar process carried along the circuit 23, 24, etc., shows that the temperature at the point 23, 24, will be 165°. This circuit here branches into two parts, one required to feed radiator No. 11 and the other to feed radiators Nos. 1, 1a.

The amount of heat given off from these two circuits is respectively found to be 28,000 and 9,000 B.T.U. By taking the temperature of the drop riser 165°, whose diameter is assumed to be 1½″, and the known surface of the two radiators 1 and 1a, their mean temperature being assumed to be 155°. The mean temperature of radiator No. 11 is assumed to be 150°.

It is therefore evident that the total 1,330 pounds of water carried by pipe 23 has to be divided in this proportion between these two sections, giving a flow respectively of 1,000 pounds and 330 in the two branches. Calculation shows that if pipe 24 is 1¼″ and 25 is 1″, the total resistance of each will be nearly the same when these respective quantities of water flow through them. These rates of flow will reduce the temperature of the return water from radiator No. 10 to 130°, while that from the other radiators will be at 140°. This is a difference which is likely to occur, and cannot be reduced without increasing the size of a portion of pipe 25 to 1¼″. As this is very undesirable, the size 1″ is accepted, and this difference is tolerated. The return pipe being horizontal the hot water is not likely to back up it and spoil the circulation, and since both radiators are at the same level no difference of back pressure can arise to stop either circulation.

A further calculation on the same lines shows that the temperature of the return on entering the boiler will be 133°, and the mean temperature of radiator II. will probably be 120° or 125°.

The total temperature difference in this circuit is therefore provisionally fixed at about 47°.

It is now necessary to calculate the total circulating power in both these circuits.

As the boiler chamber is shallow, and the radiators are not at a great height above the boiler, it is evident that if we disregard the loss of heat in the overhead pipes, the circulating power which we calculate on will be much less than the actual circulating power, and as there is no circulating power to spare, it is desirable to make a complete calculation, by estimating the mean temperature in each vertical section of the pipe.

Consider first section 1. The pipe rises to a height of 35 feet, and its mean temperature will be about 175° at a conservative estimate.

The approximate temperature of the vertical portion of 2 is 172°. The portion of the riser marked 3 is 164°, and the other portions show temperatures as table.

By the use of the formula $\Sigma H D$, the total circulating head, therefore, round to the far end is found to be 22.3 pounds per square foot, or 4.4″ of water.

Similarly, the circulating head through circuits 31—35, and 3, 4, 5, 6, and 18, 19, are found.

It is now necessary to calculate the total resistance set up in each circulation when the given quantities of water are passing through them. The way in which this is done will be understood from the table on page 525 without further explanation.

The lengths of pipe are determined, and by the addition of the figures given on page 505, the equivalent lengths of straight pipe are found, which would give the same resistance as the actual pipes. These are set out on a copy of the form, on page 525.

On adding up all these resistances it is found that the total resistance in circuit 4, 5, is 3.4″ of water. There is therefore more circulating power than necessary, which we proceed to use up by reducing the diameter of some portion of the pipe.

Now the pipe marked 5 is the most desirable to be reduced in diameter because it is more conspicuous than the other pipes. This pipe is therefore marked down to 1½″. This gives altogether a total resistance of 3.9 inches of water.

It is therefore evident that on this circuit we can reckon on an increased flow, which may be an advantage in this case.

Assuming that the resistance generated is increased uniformly throughout the entire circuit, it will be seen from an inspection of the diagram that an increase of 7% in the circulation will produce about the increased resistance that we require. This will cause a slight drop in the circulating power obtainable, and therefore an approximate estimate of the increase leads us to determine the probable water flowing through as 1,430 pounds per hour.

The calculations are now corrected to suit the increased flow, and it is found in precisely the same way as before, that we obtain a circulating force of 4″ of water, and a total resistance of 3.96. These figures are as near as it is advisable to carry the calculations, and are probably considerably more correct than our data, and therefore the calculation may safely be left at that.

This process determines finally the amount of water we are reckoning on, and it is now necessary to adjust the sizes of the circuit 31 to 35, to suit.

The total circulating power in that circuit is found to be 4.2 inches of water, and the loss of head in the sections 1, 2, 11, 17, etc., which are common both to this circulation and the other, is 1.57 inches of water.

It is thus seen that in the circuit 31 to 35 we must arrange the diameter so that 2.63 inches of water are consumed. As the portion 35 is fixed at 2″, it will be seen that the portion 31 to 34, if fixed at 1″ would not have high enough resistance.

If, therefore, the portion marked 34 is fixed at ¾″ it will be sufficient for the purpose, and it will enable this portion of the pipe to be concealed under a step which is found at that point and will avoid the construction of a channel.

The part 31, being 1″, will give about the required resistance in that pipe. The size of the branches to the radiators 7 and 8, provided they are large enough, is not of any great importance.

¾″ is considered to be large enough for 8, and the same for No. 7—a larger radiator though the connections are shorter. To ensure that this is so radiator No. 7 will require an amount of water 270 pounds, and the circulating pressure through that radiator will be about 0.12 inches of water. The radiator being 38″ high, the column outside it being at a temperature of 170°, and the temperature of the radiator at 160°.

¾″ is thus seen to be sufficient for radiator No. 7.

The above examples show the general nature of the calculations which have to be made in the frequent cases of irregular and unusual installations. It will be seen that when the data are the sizes of the radiators the calculations are very much more difficult and complicated than is the case where the B.T.U. at each point can be determined.

It is not possible to lay down general rules for the solution of these problems. Common sense and a thorough understanding of the principles involved will enable any system to be correctly sized by the process of trial and error.

CHAPTER XXVI.

ON SIZES OF PIPES FOR SYSTEMS OF ACCELERATED CIRCULATION.

THE method of calculation of pipe sizes for systems of accelerated circulation is essentially the same in principle as that which has already been dealt with for low pressure hot water circulation. The application is much easier on account of the fact that the increased circulating head which these systems provide renders of less account the variations in circulating head which are brought about by the varying densities of water as in the ordinary low pressure system. In the case of ordinary low pressure hot water heating the latter constitutes the whole motive force.

It is, however, impossible to give any *accurate* general rules for the sizing of a system of pipes on any system of accelerated circulation, unless the method according to which pipes are fixed is taken into account.

The method of fixing the pipes either increases or decreases the total head available for circulating the water through each circuit pipe. The actual head forcing water through a set of circulating pipes in which the water is forcibly circulated in a direction contrary to that in which it would circulate on the ordinary low pressure system, is essentially different from the available circulating head which is produced by the same external motive power when the design of the pipes is such that the natural circulating power produced by the difference of density in the water is in the same direction as the accelerating force.

In cases where the total head produced is very high, as, for instance, in the case of an ordinary centrifugal pump, the relatively small differences of circulating pressure produced by variations in the density of the water are relatively of small importance and may commonly be neglected, if the sizes of pipes are so arranged that a large proportion of the head generated by the accelerating system (whatever that may be) is brought to bear on the actual risers in which the contrary head or back pressure is generated.

Thus, for instance, a system of accelerated circulation is provided, such that there is a total available circulating power of, say, 10 feet of water pressure or 120 inches. If a large part of this 120″ total circulating pressure is consumed in maintaining a high velocity in relatively small main pipes, there may be so little difference of pressure at the far end of the system between the flow pipe and the return pipe that the difference of head induced by a high riser bears a relatively large proportion to this surplus balance of pressure. When this is the case, the distribution of the water between the various radiators is of necessity irregular, unless the pipes have been calculated in accordance with the principles already laid down for the sizing of a low pressure system.

The calculation of the sizes of pipes at the remote ends of a circuit needs very great care, and it is not possible to rely on approximate methods of obtaining these sizes without going into the matter in some detail.

Approximate Methods of Determining Sizes of Pipes.

Whatever system is adopted it is absolutely necessary that the total available head for forcing the water through the pipes should be known with some accuracy. This head being known, the determination must be made of the proportion of this total head which it is desirable to neutralise by friction in the main pipes, and what proportion of the same is to be left available for forcing the water through the pipes at the remote end of the system.

It is to be noted, too, that if the velocity of the water in any pipe exceeds a certain very moderate maximum which may be placed at from 30″ to 36″ per second, the flow of water through the pipes will give rise to an audible noise, which is exceedingly annoying if the room is to be used for purposes where a degree of quiet is desirable, such as in sitting-rooms, churches, and other similar buildings. In buildings such as factories the small noise which this flow of water through the pipes makes is not of considerable importance.

In determining on the proportion between the total head and the head to be consumed in the circuit pipes, it is advisable that the available head at the remote ends of the circulation shall be at least three times the amount of head produced by the natural difference of density.

In order, therefore, to make an approximate determination of these sizes it is necessary to measure the total length of the flow and return pipes, and determine what velocity of water is allowable in each of those pipes in order that the total head consumed in friction may not exceed the proportion of the total available circulating power, which it is desired to consume in the mains.

Consider, for instance, a system having a total available circulating power of 84″ of water. In this case it is desirable, for instance, that 2 feet of water pressure may be available to force the water into the last riser connected with the most remote radiators in the installation. In this case 5 feet head may be consumed in the entire length of main pipe.

It is necessary to make an estimate of the mean velocity of water in the main pipe which would correspond to a total consumption of head of 5 feet, and this mean velocity having been determined, the diagrams, Figs. 86, 87, give at once the sizes of pipe corresponding to any given number of B.T.U. which the pipe may have to carry. A still better though slightly more difficult method consists in determining the greatest mean allowable loss of head per 10 foot length in the main, and taking the corresponding capacity of pipe from the table.

The length of the circuit pipe in a similar way may be subdivided in such a way that 2 feet of water pressure may be consumed in the requisite flow of water to the most remote or index radiator in the entire installation.

If the size of the individual circuit pipe to any intermediate radiator is made too large, it is possible that the flow of water induced through that pipe will transmit so much back pressure to the return pipe that the pipes further along will not receive their due share of circulating pressure.

It is therefore important that all connections to the intermediate radiators shall be made no larger than necessary to transmit the necessary quantity of water to those radiators.

There are further practical considerations to be borne in mind relating to the possibility of very small pipes being blocked up by an accumulation of dirt, although in a carefully fitted installation all pipes are thoroughly inspected before being inserted in a job, so that they do not contain any dirt which can possibly be avoided. Yet it is not possible, however great care is taken, to ensure that particles of scale or rust should not detach themselves from the interior of the pipe, and that no lumps of red lead or hemp, or other materials used for jointing shall be accidentally introduced into the interior of the pipe system.

There is, therefore, always a liability of dirt of various kinds finding its way into the expansion tank and so into the system. Once any particles of dirt are introduced into a system of this kind it is certain that they will be carried forward by the rush of water.

If in the first place such particles of dirt enter the flow pipe, the rush of water invariably carries them in the direction of smaller and smaller pipes, until the smallest pipes in the entire system, which are generally the pipes connecting to the radiators, are reached. Such particles commonly adhere either to some ledge, such as is formed when a pipe is reduced in size, or in an elbow or bend at the junction to the radiators, or in some other similar recess in the installation.

When a piece of debris of any kind, particularly if it is of a stringy nature, such as a piece of hemp or tow or waste, so attaches itself to a projection, it at once increases the probability that future pieces of dirt passing along the same pipe will attach themselves to the mass already collected there, so that the liability to accumulation of dirt at one particular point is progressive until finally the pipe becomes absolutely blocked.

It is therefore of very great importance that in all systems of accelerated circulation special care should be taken to reamer out all pipes and so to fix them that there shall be as few interior projections or as few interruptions of smoothness of the interior walls of such pipes as possible, and further, that at all points where such accumulations can collect it shall be possible without a great deal of trouble to remove them.

The exact method of calculating the pipe sizes is on the same lines as that already explained for the low pressure system. The quantity of heat carried by each pipe should be calculated by the method already explained, as far as the pump or other accelerator.

The quantity of water required to be passed by every pipe in the system must then be calculated, proceeding outwards from the pump, these quantities being sub-divided between the various branches in the same manner as already described.

Each circulation into which the whole system can be divided must then be analysed, and each pipe length numbered as before, and the total difference of pressure at each junction must be calculated, when the assumed quantities of water pass through each pipe.

In making these somewhat extensive calculations, much time can be saved by dividing the main pipe into groups as far as each important branch and figuring on plan the difference of pressure between the flow and return pipe at each junction. Each junction then becomes the centre from which the loss of head round each secondary circulation can be calculated.

The loss of head from flow and return along each such secondary circulation should be the same. If this is not done it is certain that those circulations having too small resistances will pass more than their required quantity of water and the balance throughout the system will be spoilt.

It is to be noted that any alteration in the circulation after the calculations are made will of necessity vitiate to some extent the whole of the calculations previously made. It is therefore necessary in order to save calculation that no important alterations in the job shall be made without consulting the person who has made the calculations.

It is always advisable for all systems of accelerated circulation to provide each radiator with valves having a double adjustment, so as to compensate for irregularities in the pipe which, as already explained, always exist. But in making the calculations the existence of these adjustable valves should not be allowed for to any greater extent than the difference between two successive sizes in any one section of the circulation, that is to say, the circulation should be arranged in such a way that the total loss of head from flow pipe to return pipe at any one junction should be as nearly as near to the calculated available head as can be given by altering the size of any one pipe.

A permanently uniform result can be secured with much greater certainty by properly calculating the sizes of the pipes, than by adjusting a valve. Adjustments made by valves are apt to be uncertain and variable and the valves are apt to be tampered with after the adjustments are made.

In the opinion of the present writer, it is not generally necessary or desirable to sub-divide any but a long pipe into two sections of different sizes merely in order to bring the calculated values to an exact figure. The difference, however, between the calculated loss of head and the exact available head should not be greater than 5%. If greater differences than this are allowed to creep into the calculations, the whole of the calculation will be rendered uncertain.

Within this figure the proportionate distribution of water between the various branches may be relied on, unless some extraneous influence, such as the accumulation of air or debris in any portion of the pipe interferes with the free flow.

If the calculations are kept within this degree the quantities of water flowing through all sections will be approximately correct. Though they may vary up to 10%, or even 15%, from the calculated values, they will vary in proportion. But even a variation of this magnitude does not produce any very great variation in the heat emitted from any particular radiator, as may be seen by examining the curves in Fig. 88.

In all cases the addition to or reduction of the available circulating power in each circuit owing to difference in density in the water flowing to and from the various radiators and coils should be made. Suppose, for instance, the available circulating pressure at a junction A is found to be, say, 17″ of water pressure, and on one circuit the calculation on the natural basis, there would be a positive circulating force of, say, 4.5″ of water. The total circulating head for that circuit should be calculated as 17 + 4.5.

If in another circulation a contrary or opposing pressure of, say, 3.2″ of water pressure would be produced by the alteration in the density of water, the total available circulating power to be calculated for that circuit should be 17 — 3.2″ of water pressure.

If these variations are not taken account of, they will themselves produce variation of head, which will be sufficient to interfere seriously with the calculated flow of water. The greater the available head the less important do such variations become, although in all cases it is advisable to take account of them if an accurate result is desired.

It is very certain that when a system is finally fixed, the flow of water which will actually take place along any given pipe length will be determined not only by the resistance of that pipe itself, but by the resistance of all other circulations connected to the same point. It is essentially true that the total friction generated in any two branches connected in parallel on the mains must be the same, and the forces of nature will so adjust the flows of water that the total loss of head is accurately the same in each such branch circulation.

If the resistance of the pipes, therefore, can be accurately known, the flow of water in each section can be accurately determined. It will be seen from a consideration of the previous discussion that the effect of any alteration in the resistance of any part of the circuit will be felt throughout the entire system.

It may be taken as quite certain that nature's solution of the multitudes of simultaneous equations involved in each system will be rigidly accurate, and the necessity for the engineer is to make the solution of his own equations just sufficiently precise that any inevitable variation from accuracy which nature may detect throughout the whole system will not be so great as to upset the ultimate result which the engineer seeks, which is the delivery of certain minimum quantities of heat at the given points.

The amount of water flowing through the radiators does not, fortunately, diminish the output of heat of those radiators in the same proportion as the flow of water. If it did, it is very certain that every installation, however carefully it might have been designed, would show wide variations from accuracy.

It has been proved in previous chapters of this work that not only are the amounts of heat required to produce certain results in different rooms subject to very considerable uncertainties.

The fundamental condition, therefore, which the engineer aims to produce is that whatever inaccuracies there may have been in his own calculations of heat loss, or of flow of water in the pipes, that all taken together, the amounts of heat delivered to those points shall not be below those actually required when all variations from theoretical accuracy have been taken into account.

For instance, suppose that W is the amount of heat *actually* required at a certain point in an installation to produce the guaranteed results, and that the amount of heat *calculated* to be required by the engineer is W_1. Assume also that the engineer's calculations give the temperature of the water flowing *to* the radiator as T_1, *from* the radiator as t_1, and its quantity as Q_1 pounds per hour, and assume that the *actual* temperature in the flow water is T, the actual quantity of water Q, and that the outlet temperature in these conditions is t. The amount of heat which will actually be delivered in these circumstances is

$$Q\,(T - t)$$

The engineer's fundamental aim is (1) that this quantity should at any rate not be less that W, and (2) that it should be as little greater than W as possible.

The value

$$Q\,(T - t)$$

may also be expressed as

$$A\,K\,(\frac{T + t}{2} - t_a)$$

In order that the radiator whose area is A, may give off the requisite quantity of heat, the necessary condition is that $\frac{T + t}{2}$ shall be so great that the value of the expression

$$A\,K\,(\frac{T + t}{2} - t_a)$$

shall be at least equal to W. In order that this may be so the quantity $Q\,(T - t)$ must at least be equal to W, whatever the value of $Q\;T\;t$ may be.

If Q is too small, owing to some short circuiting in the system of pipes, then whatever value T may have the value of t will fall so low that

$$\left(\frac{T + t}{2}\right)$$

has not the desired value.

The manner in which the value of

$$(\frac{T + t}{2} - t_a)$$

varies when t is reduced, is seen very clearly in the diagram, Fig. 88.

The reduction below the calculated value of Q only produces a relatively small alteration in the value of T, but it produces a large alteration in the value of t.

Now the value of the surface A may be made great enough to counteract the deficiency in the value of t. To increase the value of A by increasing the size of the radiator has the effect of reducing the value of t.

The quantity of water Q is determined in the main in accelerated circulations by the general design of the pipes. Therefore an increase in the value of A produces an increase in the value W of the heat given off in direct proportion as it increases the value $(T - t)$.

When Q is constant the increase in the value of A decreases the mean temperature of the radiator, but although this is so, and although it necessarily produces an increase in the value W, that increase is not in proportion to the increased value of A.

It is to be noted that in systems of accelerated circulation a decrease in the value of t only produces a proportionately slight increase in the circulating power, and that only in those cases where the pipe is fixed in accordance with the principles of low pressure hot water circulation.

It is quite frequently the case in systems of accelerated circulation, that pipes are fixed in a contrary sense, then the decrease in the value of t consequent on an increase in the size of the radiator *reduces* the circulating power, and to the same extent reduces the quantity of water flowing through the radiator, so that in an installation of this character it depends on other considerations whether it is of any use to increase the size of the radiator in order to provide more heat.

It is always of use to increase the flow of water through the pipe whether or not the radiator is increased in size, although a mere increase in the supply of water, unless this is *very* deficient, does not produce a very great increase in heat emission, as will be seen from a consideration of the diagram, Fig. 88.

These somewhat complicated considerations should be thoroughly mastered by anyone who desires to understand the exceedingly interesting problems which arise in connection with systems of heating by accelerated circulation.

CHAPTER XXVII.

ON HEATING BY HIGH PRESSURE HOT WATER.

THE principle on which the high pressure heating systems work is identical with that of the ordinary low pressure. The circulating power is derived from the difference in weight of a hot column of water, probably mixed with steam bubbles, and a cold column of water in the return pipe. But the circulating power obtained on a high pressure system is so much greater than on a low pressure system, that the rules which govern the fixing of the circulating pipes in the latter need not in all cases be adhered to, especially considering the greater simplicity of the circulation in the high pressure system.

The name " high pressure " is given to this method of circulating the water because the system is closed out of contact with the atmosphere, by means of a loaded safety valve, so that the pressure within the pipes can be raised much higher than is the case in the low pressure system, as water, of course, does not turn into steam at so low a temperature, when the pressure is increased. But on account of the existence of this pressure, the danger of an explosion is distinctly increased, although with a well constructed high pressure apparatus an explosion is an incident of extreme rarity.

The form of heating surface in the high pressure system is in practically all cases a single hydraulic pipe of much greater strength than the ordinary pipes used for low pressure work, and the pipe is continuous from end to end. The boiler consists of a coil of this same pipe wrapped round and enclosed within the furnace. The interior diameter of the pipe is in all cases about 0.875″, the outside diameter being about 1.318″, the surface of one foot run being about 0.345 sq. foot, or 2.9 ft. run in one square foot.

Since the temperatures attained in this pipe are so much higher than in a low pressure pipe, the amount of heat obtained from one foot is very much larger, and therefore the system is on the whole economical to carry out, though it is not economical in upkeep, and further, it gives rise to a kind of heat which many people have a great objection to, owing to the burnt effect on the dust particles of the air.

The system consists of an endless coil of pipes, one portion of which is convoluted and placed in the furnace. Where much heat emission is required within the chamber to be warmed the coil is also convoluted, so as to include in the chamber the amount of heating surface necessary.

General Remarks.

Before being used all high pressure hot water systems ought to be tested to a pressure of about half a ton per square inch at least, and care must be taken that it is completely filled with water, and that no air is left in the pipes. Great trouble is frequently caused in high pressure systems because of air left in the pipes.

The air has, in most cases, to be ejected by means of a pump, which forces the water into the pipe, thereby carrying the air before it. Great care must also be taken to see that no dirt is left in the pipes.

Trouble is frequently experienced with high pressure systems if one circuit is branched into two or more, owing to short circuiting, and only engineers specially experienced in this form of heating should undertake the construction of branched high pressure circuits. On this system the pipes near the boiler can frequently be raised to a blue heat, and it is commonly employed for raising temperatures in baking and enamelling ovens, where temperatures of 500° and 600° are necessary.

The pipe joints are made, not as in low pressure work, but by means of a special hydraulic joint, whereby the end of one of the pipes is sharpened, and the other pipe being left flat, the two ends being drawn together by right and left-handed screw threads. The sharp edge is jammed into the relatively soft metal and a metal to metal joint is thus made without the employment of red lead or other jointing material.

The upper part of the system is connected with a strong expansion tank, which is loaded with a safety valve, according to the pressure at which the system is to be worked. For very high pressures the expansion tank consists of a suitable number of vertical pipes whose contents are about half of the whole contents of the apparatus.

In common heating practice the limiting water temperature is about 300° F., corresponding to a pressure of about 75 pounds per square inch, though the safety valves are frequently loaded with much heavier pressures than these, 1,000 pounds per square inch not being uncommon. Every high pressure heating system ought to be supplied with a hydraulic pressure gauge.

Special precautions have to be taken against freezing of the pipes, which will cause the pipes to burst, however strong they may be. Sundry methods of avoiding the freezing of the water have been attempted, by such devices as a mixture of glycerine and salt, but with very little success. Alcohol has also been tried.

The drawbacks to the use of these substances are various. Glycerine evaporates through the joints and becomes charred. A mixture of salt causes corrosion of the pipes. Alcohol (generally added in the proportion of 1 of alcohol to 2 of water), owing to the high temperature, is frequently distilled off from the water into the expansion tank. Decomposition of these compounds into gases is sometimes caused by the high temperature and a high expansible pressure thereby generated which persists even when the water is cold.

There are, however, special patent mixtures which can be added to the water, which prevent freezing, and are claimed by their manufacturers to avoid the disadvantages of salt and other preparations.

The great advantage of the system is that it overcomes difficulties of levels, and on account of the small water contents of the pipes, the heat is very quickly raised.

The difference in temperature of the flow and return pipes in an ordinary high pressure system is very great. This is of great assistance to the circulation, but of course it makes of great importance the distance at which the heating coils are connected. The difference in heat emission between a coil near the boiler and a coil far away from it is very great. Since regulation of the individual coils is impossible on account of the great difference of temperature, the distribution of heat is not uniform at different temperatures. The system is therefore suitable chiefly for single rooms or halls in which all the heat given off passes into one room.

Calculation of Heating Surfaces.

Since there is only one kind and size of pipe the calculation of the pipe lengths is comparatively simple for plain cases, but becomes excessively complicated for branched circuits. It is best treated by dividing the entire circuit into three parts.

(1) Coil of pipes forming the boiler.

(2) Flow and return pipes, which are not required for heating surface, and which should be coated with a very efficient inorganic composition which will not char nor crack under excessive temperatures.

(3) The heating pipe itself.

The second of these lengths is fixed by the design. The other two must be calculated. The total surface of the heating coils should be calculated for the lowest outside temperature, but the size of individual coils should be proportioned for a medium temperature.

In order to calculate the quantity of pipe to be fixed in the fire, forming the boiler, it may be noted that with a furnace temperature of 1,800° F. a transmission of about 600 B.T.U. per foot run of fire coil may be reckoned on, while with a temperature of 2,200° F., 750 B.T.U. may be obtained.

The amount of water required to be circulated is given by the formula

$$\frac{W}{t_1 - t_2}$$

where t_1 is the flow temperature and t_2 the return, and W the whole heat required per hour.

Now since the interior diameter of the pipes is almost universally ⅞″, and the general flow temperature is taken as 300°, the return temperature being 175°. The density of water at 300° is 57.25 and at 175° — 60.68 lbs. per cubic foot.

The mean density of the water may be taken as the mean of the density of water between 300° and 175°, which will be about 60 lbs. per cubic foot, or 28.8 cubic inches per pound, and the mean temperature at 240° F.

The formula for the velocity can be as follows :—

$$v = \frac{28.8\ W}{(0.875)^2 \times 0.785 \times 3{,}600 \times 125} = .0001155\ W.$$

Exactly as in low pressure hot water heating, the formula for the loss of head by friction in the pipe is

$$p_1 - p_2 = \rho\ \frac{l}{d}\ \frac{v^2}{2\ g} + \Sigma\,\zeta\ \frac{v^2}{2\ g}$$

It will thus be seen that the calculations for a high pressure system where all the pipes are in one apartment are simple. Where, however, the whole pipe is divided between several rooms, the complication is great, since the temperature of the water varies so greatly. Indeed, it is by far preferable where several rooms have to be heated, to supply an independent installation for each room, each connected to its own separate boiler coil in the furnace. The difficulties of correct calculation, and above all of making the apparatus work out in practice according to the calculations, are great.

Calculations.

For rough calculations it will be sufficient to reckon the value of K as being 2.7. The length of pipe may be determined according to the formula

$$W = 2.7 \times .345 \times l \times 180 \text{ or } l = \frac{W}{168}$$

and the boiler part of the coil from the formula

$$l = \frac{W}{600}$$

for a powerful boiler, and for a medium power

$$l = \frac{W}{750}$$

Jones, in "Heating by Hot Water," gives the correct proportion between the boiler and the heating pipes as varying from 1/6 to 1/10, according to the temperature required.

In simple cases this calculation is probably as exact as it is necessary to get, for with this system, if slightly too little pipe has been allowed, it is always possible to raise the temperature to make up for the deficiency, so long, of course, as the maximum admissible temperature is not exceeded.

In cases where the system is complicated by separating a single coil into several different parts, each heating different rooms, it is only possible to make the calculation for one particular temperature, and for any other temperature there will be relative inaccuracy, that is, the same relative amounts of heat will not be supplied to the different rooms. In this case it is necessary to add together the total quantities of heat required in all the rooms, and determine thereby the total length of pipe required as in the first case. It is the sub-division of this total length of pipe which constitutes the difficulty, and it is probable that this can best be effected by a graphical construction.

The analytical solution of the problem is very difficult and complicated, and the result is probably no nearer than will be obtained by the method now about to be described.

That method consists of plotting, by the construction already explained at curve V on page 277, the total loss of heat from the whole length of pipe, on a base representing the length of pipe itself, the value of K being taken as 2.7.

If then the total height of this curve between the two points representing the total length of pipe is divided into the same proportion as the respective quantities of heat, the abscissa will show the corresponding length of pipe which should be fixed in each room.

CHAPTER XXVIII.

FLOW OF STEAM IN PIPES.

THE general theory governing the flow of steam in pipes is identical with that which has been already dealt with for fluids in general, but in the case of steam there are other important factors to be taken into consideration, which do not exist in the case of water and air.

In the case of air, so far as the heating engineer has to deal with it, the pressure, and therefore the density of the air in the same flue never vary considerably. The greatest difference of pressure which in practice ever causes or is caused by the movement of air along a flue for ventilating purposes does not exceed one or two inches of water. The absolute pressure of the air is between 350 and 400 inches of water. For purposes with which the heating engineer has to do the difference of a fraction of 1% can be disregarded.

Similarly with water, however great the difference of pressure may be, it does not produce a substantial alteration in the density of the fluid. It is true that alterations in density of appreciable magnitude are produced by differences of temperature in the water, but the whole expansion of water between 32° and 212° is below 5%. This difference is not great enough to cause any considerable difficulty in dealing with the resulting change of density by an approximation according to which the mean density of the water in the pipe is calculated as the mean of the densities at the beginning and end of the pipe.

This does not introduce any serious complication or inaccuracy into the equations.

With steam, however, the case is quite otherwise. Not only do differences of pressure produce relatively large differences of density in the same pipe line, but also the important question must be considered of the amount of condensation of steam in a length of pipe. In many cases this bears an appreciable proportion to the total quantity of steam passing along the pipe. There is the still further complication that the condensation not only reduces the volume of steam passing along the pipe from point to point, but also produces water in the pipe itself. This has an important effect on the flow of steam, and on the co-efficient of friction. Therefore, though the laws relating to the flow of steam are fundamentally the same as those relating to the flow of other fluids, they become greatly modified by the introduction of these extra considerations which do not exist in the case of the other fluids with which we have to deal. The treatment of steam is therefore more difficult and complicated than in the cases previously given.

In this investigation, due to Rietschel following Fischer, the symbols below will be employed :—

W The quantity of heat in calories which can be delivered through the pipe at the remote end in one hour.

W' The quantity of heat in calories lost in one hour from the surface of the pipe.

Q The quantity of steam in kilos per hour delivered at the end of the pipe.

Q' The quantity of steam in kilos condensed per hour through loss of heat in the same pipe.

p_1 The absolute pressure of the steam at the end of the pipe in kilos per square metre.

p_2 The absolute pressure of steam at the beginning of the pipe.

l The length of the pipe in metres.

d The inner diameter of the pipe in metres.

D The outer diameter in metres.

t_m The temperature corresponding to the mean pressure of the steam in the pipe in deg. C.

t The temperature of the air surrounding the pipe in deg. C.

λ_m The latent heat of steam at the mean pressure.

k The transmission co-efficient calculated with regard to the outer area of the pipe (whether coated or not).

Consider any point in the pipe at a distance x from the end of the pipe. The quantity of steam condensed by this last section of pipe x feet long is

$$\frac{D \pi (t_m - t) K x}{\lambda_m}$$

Now since at the end of the pipe a quantity Q pounds of steam is delivered per hour, the total quantity of steam passing the chosen point must be

$$Q + \frac{D \pi (t_m - t) K x}{\lambda_m}$$

Therefore the velocity of the steam at that point must be

$$v = \frac{Q + \dfrac{D\,\pi\,(t_m - t)\,K\,x}{\lambda_m}}{3{,}600\gamma\,\dfrac{d^2\,\pi}{4}} \text{ metres per sec.}$$

Now, since the flow of steam in the pipe, apart from the extra considerations to which attention has already been drawn, is governed by the same laws as in the case of water and air, previously considered, it is evident that if we consider the loss of head in an extremely small length of the pipe $d\,x$, the loss of heat from it may be disregarded, as it bears an infinitely small relation to the total steam flowing through. The difference of pressure ($d\,p$) at the ends of this small length will be governed by the same law as in the case of water and air, that is

$$d\,p = \frac{v^2}{2\,g}\,\gamma\,\frac{\rho}{d}\,d\,x$$

This equation will hold for every such small length.

Now substituting in this equation the value of v given above, we have

$$d\,p = \frac{\left(Q + \dfrac{D\,\pi\,(t_m - t)\,K\,x}{\lambda_m}\right)^2 \gamma\,\rho}{2\,g\,\left(3{,}600\,\gamma\,\dfrac{d^2\,\pi}{4}\right)^2 d}\,d\,x$$

In order to shorten this somewhat cumbersome expression, take out of the equation all the constants, which will, of course, always reappear as constants in the future equations, and therefore in order to avoid writing them all, put

$$\frac{4^2\,\rho}{2\,g\,.3600^2\,\pi^2} = A$$

$$\frac{\pi\,(t_m - t)\,K}{\lambda_m} = B$$

The equation thus becomes

$$d\,p\,\gamma = \frac{A}{d^5}\,(Q + D\,B\,x)^2\,d\,x$$

Now the value of the density of steam can be written

$$\gamma = a + b\,p$$

in which the values in C.G.S. units are (Rietschel)

$$a = 0.1521$$
$$b = 0.0000498$$

This equation is approximate only, the errors from absolute accuracy being about 1.5%.

Hence we have

$$(a + b\,p)\,d\,p = \frac{A}{d^5}(Q + D\,B\,x)^2\,d\,x$$

The integration of this equation, and the substitution of their proper values for A and B give the following result, where

$$\frac{\rho}{2\,g} = 0.00105$$

a value determined by the recent experiments of Eberla :—

$$p_2{}^1 = \sqrt{\frac{17{,}610\,l}{(100\,d)^5}\left\{3\,Q^2 + Q^1\,(3\,Q + Q^1)\right\} + (p_1 + 3{,}060)^2} - 3{,}060$$

This equation would be somewhat complicated if the single resistances ζ were separately considered. It only takes account of the resistances in straight lengths of pipe. Each single resistance would produce a difference of pressure of

$$\frac{v^2}{2\,g}\,\gamma\,\zeta$$

and the varying values of γ would cause considerable complication.

The alterations of pressure produced by these single resistances would in any case be small. In order that any error may be on the right side, the velocity is taken as the velocity at the beginning of the pipe, and the density as that at the end of the pipe, so that we have for the maximum total loss of pressure due to all single resistances as follows :—

$$p_2{}'' = \frac{v_a{}^2}{2\,g}\,\gamma\,\Sigma\,\zeta$$

where v_a = velocity at beginning of pipe = $\dfrac{Q + Q^1}{3{,}600\,\gamma\,\dfrac{d^2\,\pi}{4}}$

γ = density at end of pipe.

Whence

$$p_2{}'' = \frac{(Q + Q_1)^2\,\Sigma\,\zeta}{156{,}854{,}070\cdot\gamma\,d^4} = \frac{(Q + Q_1)^2\,\Sigma\,\zeta}{(111.9\,d)^4\,\gamma}$$

The values of ζ are to be taken as follows :—

A right angled elbow	1
A bend	0.3 to 0.5
An open valve	0.5 to 1
An open cock	0.3 to 1
A bend of more than 5 diameters radius ...	— 0

Hence we have, allowing for heat loss, friction, and single resistances, the following equation, which may be simplified as follows :—

$$p_2 = p_2' + p_2''$$

$$= \sqrt{\frac{17{,}610\,l}{(100\,d)^5}\left\{3\,Q^2 + Q^1\,(3\,Q + Q^1)\right\} + (p_1 + 3{,}060)^2 - 3{,}060 + \frac{(Q + Q^1)^2\,\Sigma\,\zeta}{(111.9\,d)^4\,\gamma}}$$

Explanation of Diagrams.

It is evident that such an equation as this is considerably too complicated for general use, and in order that it may be used in practice the curves shown on Figs. 133 and 134 have been calculated and plotted. The simplification of the equation is given fully in Rietschel's classical work (Springer, Berlin).

The amount of steam delivered by a pipe of any given size varies, of course, with the pressure. For each pipe the drop in pressure increases with the amount of steam delivered, as will be evident from the equation.

It has therefore been thought advisable to plot a set of curves for each size of pipe in common use for various pressures, so that the corresponding drop in pressure at any pressure and delivery can be at once determined without calculation.

In each case the curve is labelled with a figure which represents the mean of the initial and final pressures in pounds per square inch. In order clearly to label the curves alternate sizes have been plotted on separate diagrams to avoid confusion.

In cases where it is required to find the drop in pressure corresponding to a mean pressure in the pipe between two of the values for which curves are given, a corresponding interpolation will give the required figure.

For convenience of use the diagrams have been plotted in two different ways. Where high pressure steam is employed it is generally necessary to know the number of units of heat that will be delivered at the end of a pipe of the given length, corresponding to a certain drop in pressure between the boiler and the end where the steam is delivered. In this case the drop in pressure is required in pounds per square inch.

The table takes account of the steam condensed in the pipe in the way thither, the delivery being given as the nett number of pounds of steam per hour delivered according to the above formula at the end of the pipe.

The requirements of low pressure calculations are somewhat different. In this case it is generally required to know the number of B.T.U. which can be delivered at the end of a pipe by a given drop in pressure, but the pressure drop which can be allowed in this case is, of course, very much smaller than in the case of high pressure steam.

Fundamentally, of course, these requirements are the same. Any number of B.T.U. can be at once reduced to pounds of steam by the mere process of dividing by the latent heat, and inches of water column can with equal ease be reduced to pounds per square inch.

For low pressure steam, however, a unit of one pound per square inch as the unit of pressure is a great deal too high. The maximum pressure for which such installations are generally calculated should not exceed 2 or 3 pounds per square inch, and the installations should generally work at as low a boiler pressure as ¾ of a pound per square inch, the steam being uniformly distributed over the radiators at that pressure.

If, therefore, the unit of pressure is taken as one pound per square inch, the individual drops through short lengths of pipe will be represented by microscopic fractions, which is very undesirable, as it tends to give a sense of unreality to the calculations. The toleration of the ounce as a unit should be rigidly refused by all scientific persons on account of its inconvenience.

It has therefore been thought advisable in the present work to give the pressures for low pressure calculations in inches of water column. The temperature of the water column in which the calculations are made is taken as about 200° at which the density is 60 pounds per cubic foot, so that if in any given case it is possible or necessary to make actual measurements by means of a water gauge, no corrections need be made to the gauge readings to compensate for the difference in temperature.

At this density 1 inch of water column = 5 lb. per sq. foot (see page 480).

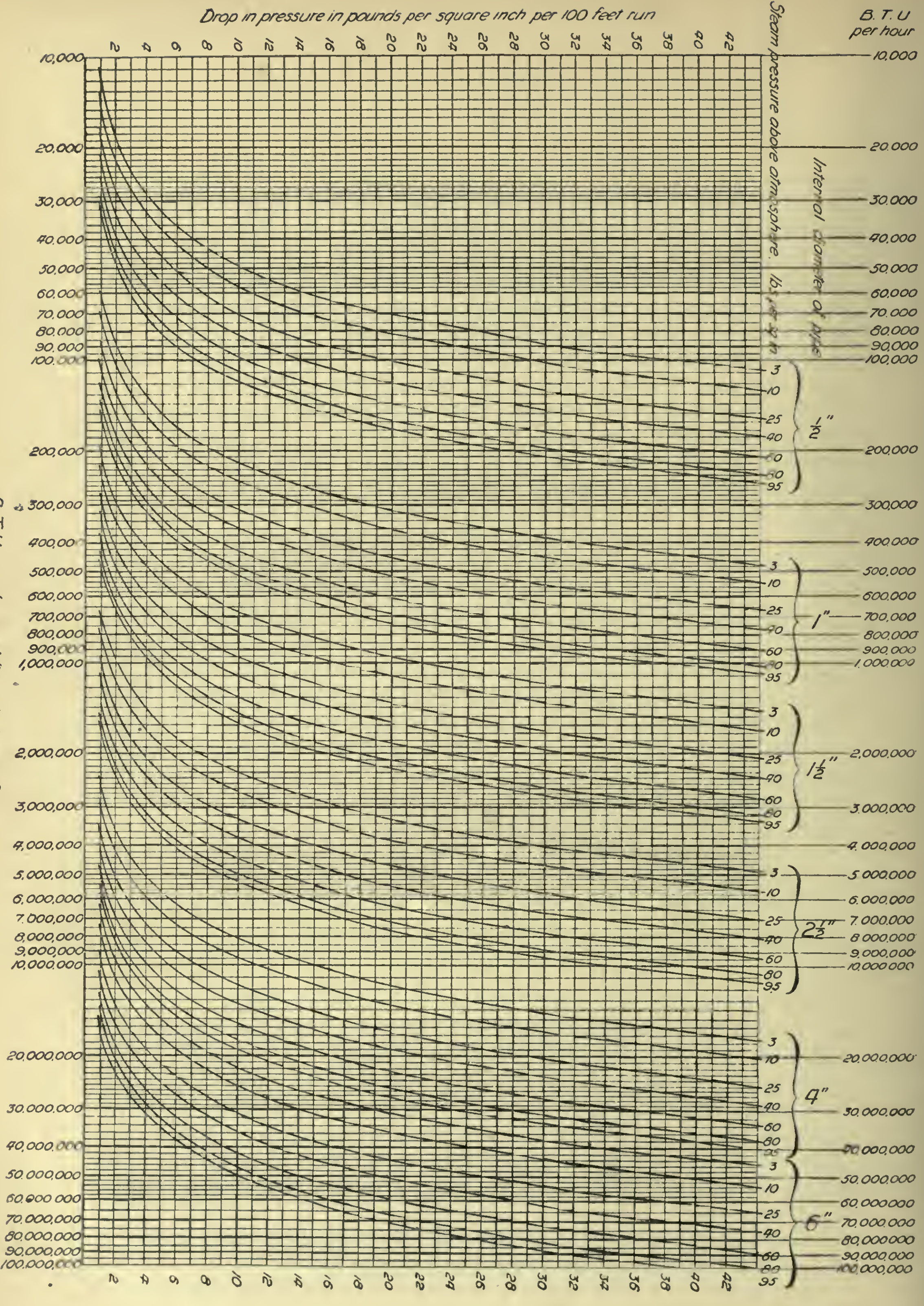
The second set of diagrams, Figs. 137 and 138, are calculated to give deliveries in B.T.U. and drops of pressure in inches of water, both for bare pipe and when the pipe is coated with a 60% quality of covering.

The differences in friction for small differences of initial pressure are not very great, although, of course, considerably greater drops can be allowed when the initial pressure is, say, 3 pounds per square inch, than when it is, say, 0.5 pounds per square inch.

It is assumed in all these tables that proper provision has been made for draining off the condensed water. According to the perfection of the provision made, so the drop in pressure is reduced. It is evident that great accuracy is not to be expected since the result depends, as in so many previous cases, on factors the exact value of which cannot possibly be determined.

Calculation of sizes of distributing pipes, where these are required to distribute different quantities of heat to different points.

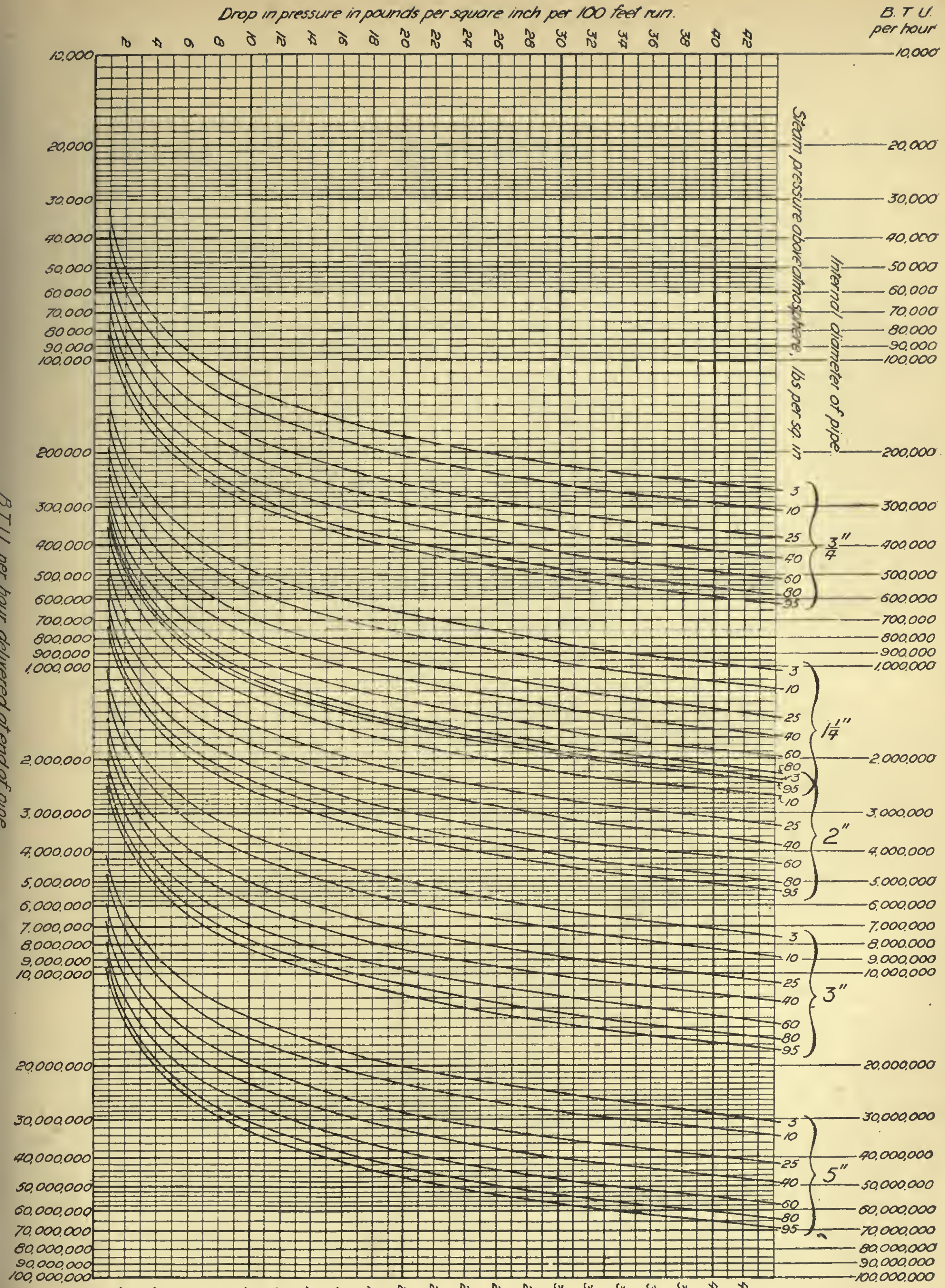
In the case of high pressure steam, exact calculations of the pressure



DELIVERY OF STEAM IN HIGH PRESSURE STEAM PIPES

A.H. Barker, delt.

FIG. 133.



DELIVERY OF STEAM IN HIGH PRESSURE STEAM PIPES

A.H. Barker, delt.

FIG. 134.

at points of delivery are not generally of very considerable importance. Where steam is required to provide motive power for engines, it is however necessary that the pressure shall not fall below a certain point, and the pipe shall be sufficiently large that there are no considerable periodic fluctuations owing to the differences in the supply during one revolution.

It is, of course, inevitable that however constant the pressure of the supply may be, there will be fluctuations at the end of a long pipe when the regular demand for steam varies from any cause. This can only be avoided by raising a higher pressure than it is required to use and employing a reducing valve near the point at which the steam is used.

The amount of fluctuation then depends on the perfection of the reducing valve. But if the pressure at some distance from the point is kept constant, fluctuations at the point itself cannot be avoided by any other means.

When high pressure steam is employed for such purposes as heating water or for steam heating at reduced pressures after passing through a reducing valve, the exact value of the terminal pressure is not of great importance, provided it is kept within reasonable limits.

The power of a water heater is commonly not and probably cannot be calculated very closely. A considerable margin of power is, or should be, provided, so that if a heater is designed for use at 50 pounds per square inch, it will probably in most cases be powerful enough if it is used at 30 or 40.

In fixing the diameter of such steam pipes, therefore, while it is desirable to be accurate and well on the right side, there is no object in making very close calculations.

The general method to be employed is the same as for low pressure steam, namely, make a complete plan of the entire system of steam pipes, figure on each end the amount of steam required at that point per hour and the desired pressure. Assume an appropriate size of pipe in each case, calculate back to the first junction for each branch what pressure will be necessary there in order that the given quantity of steam may be delivered at the desired pressure at each delivery point. Calculate the total quantity of steam passing through the main leading to the junction and carry the calculation further back to the next junction towards the boiler. Eventually, calculate what the pressure at the boiler will be under those given conditions.

If this is higher than it is intended the boiler pressure shall be, the size of pipes and the drops must be adjusted—

(1) So that when the required quantities of steam are passing through each pipe, and when adequate pressures are maintained at each terminal, the calculated pressure at each junction is the same for all pipes meeting there.

(2) So that the total pressure required at the boiler shall not be greater than required. It is very certain that the loss of head by friction in the pipe will exactly correspond to the amount of steam passing along it, and that there can be only one pressure at each junction.

Principle of calculations for low pressure steam.—These are exactly the same in principle as the preceding, but the object aimed at in designing a system of pipes for low pressure steam is that at the calculated boiler pressure, the pressure in the pipe at the inlet to each radiator *must be the same throughout the entire installation* when the given quantity of steam is passing.

It is very desirable in a low pressure installation that the pressure within the radiator shall be not greater than atmospheric. Where, however, the bottom of a radiator is only fixed a very short distance above the water level in the boiler, the difference in pressure between the interior of the radiator and the boiler must not exceed the pressure due to a water column of this height, otherwise the radiator will accumulate water.

These points are dealt with in the following chapter.

CHAPTER XXIX.

ON THE DISTRIBUTION OF HEAT BY STEAM AT HIGH PRESSURE.

HIGH PRESSURE STEAM INSTALLATIONS.

General Remarks.

IN large installations serving big institutions, steam at a greater or less pressure is frequently required for a great variety of services, from which the water of condensation cannot be returned to the boiler. In such installations, therefore, a continual supply of fresh water is necessary for feeding purposes.

In these cases the use of sectional boilers is generally undesirable on account of the impossibility of cleaning out the fur or deposit which results from the evaporation of hard water from the interior of the sections.

No entirely satisfactory method has yet been devised for removing scale from a sectional boiler. Some kinds of scale can be partly removed by a very dilute solution of hydrochloric acid, though the application of this highly corrosive substance may in some circumstances damage the interior of the boiler or the joints. Therefore, where the boiler must be fed by anything but condensed water or rain water, it is often impossible to use such boilers. A wrought iron boiler provided with manhole must then be used, the interior of which can be periodically inspected and cleaned.

These boilers are, however, invariably considerably more expensive than the cast iron boilers, and they should not be used unless the use of the latter is absolutely impossible, if economy in first cost and upkeep is, as usual, an important consideration.

Where steam is required for such purposes as cooking, laundry work, sterilization, etc., the wrought iron boiler is generally an essential on account of the high pressure which may be required.

The pressure at which such boilers should work, varies according to circumstances.

It is only for one or two purposes that really high pressure steam is necessary. Such purposes are, sterilization, and the production of motive power for electric lighting, or other similar purposes. But where high pressures are employed, the size of the pipe necessary to convey a given quantity of heat to distant points, and to use it there, is considerably reduced, owing to the increased density of high pressure steam. Thus, the pipe sizes required where the steam is generated at a considerable pressure are much less than where the steam is conveyed at a lower pressure.

But since the temperature of high pressure steam is so much greater than that of low pressure steam, there is a tendency for its production to be more expensive, owing to the greater loss of heat from all apparatus which contain it.

On the other hand, where low pressure steam is employed, larger pipes must be used, and the surface of such pipes is, of course, greater than when the pipes are small, and this tends to reduce the increased cost of the generation of high pressure steam.

Advantages and Disadvantages of Centralised Plant.

Large installations are frequently supplied in modern times with a combination of plant, which includes the steam engine for the generation of electric light and electrical power, the exhaust steam given off from the engines or turbines being employed to heat service water or buildings, or for other similar purposes. Such plant is, of course, expensive, both in first cost and upkeep.

It also needs skilled attention, but it is very convenient in operation, and when well designed and well laid down and well taken care of, it requires little in the way of repairs, and can generally be relied on not to cause interruption by breaking down.

From the point of view of economy and upkeep it is very desirable that the length of steam piping, especially high pressure steam piping, should be reduced as much as possible, that is to say, all apparatus for which pressure steam is necessary should, where it is possible, be placed in the immediate neighbourhood of the boiler, so that this loss may be as small as practicable, and further, the coating employed for pipes conveying steam should in all cases be as good as can be obtained.

Where the heat is to be widely distributed to many different points the loss of heat from the pipes conveying heat is always so relatively great that the actual fuel consumption is greater in the case of centralised plant than with a number of independently fired boilers, situated near the points at which the heat is required.

Of course, the attention necessary to a number of separate boilers is greater than if one large boiler is placed in a central situation, and the conveyance of the necessary fuel is more inconvenient. On these grounds the centralised plants have of late years become fashionable.

But it must be clearly pointed out that unless a large variety of services draw their supply from the boiler real economy cannot be claimed for such plants, especially seeing that the boiler itself is of a more expensive type, and that the attention required is also more skilled, and further, that for a continuous operation of such plant, the constant presence of the attendant is necessary, which it is not in the case of the smaller and separate installations. The boiler, too, is not generally so efficient as a good sectional boiler.

Pipe Lines for High Pressure Installations.

For very large and widely extended installations a common method of distributing heat is to generate steam from high pressure boilers at pressures varying from 50 to 150 pounds per square inch, and conducting the steam through a range of high pressure steam pipes to distant points, and there utilising the heat by various methods. For this purpose generally long horizontal ranges of pipes are necessary.

Condensed Water in Mains.

It is of the highest importance in installations of this character that these ranges of pipes shall be fixed in such a way as to clear themselves of the considerable amount of condensed water which will be formed in them. The general method by which this is done is to ensure that every part of the steam pipe shall fall gradually in the same direction as the steam flows. A common rate of fall is 1″ in ten feet run, but a greater fall than this is desirable if it can be secured. The water formed will then be carried along in the same direction as the steam, and there will be no conflict in the pipe between the water endeavouring to return and the steam flowing in the opposite direction. Such conditions invariably result in the water being caught up by the steam and hurled at a great velocity along the pipe, giving rise to loud explosive cracks, due either to the sudden condensation of quantities of steam or to the hurling of the water against the ends of the pipe. Such knocking often causes the bursting of joints, and not infrequently the rupture of the pipe itself.

In order to avoid this, the arrangement above described should be made, and when, owing to its continual fall, the pipe has reached about two feet or thereabouts from the floor of the channel in which it is being carried, a drip should be inserted in the pipe, and some form of simple water separator, water trap and steam trap provided, so that the water may be automatically removed from the pipe. Steam may by this means be carried very great distances. The cost, however, of maintaining such long ranges of pipes is great, owing to the large amount of steam condensed in them. The cost of generating this steam is frequently a large proportion of the total cost of upkeep. In one case tested by the Author the steam condensed in the mains represented 18% of the total steam generated in the boilers. The pipes were covered with non-conducting composition, the flanges being left exposed, and the pipe passed through a rather damp underground channel about 6′ high by 4′ wide.

Methods of Utilisation of High Pressure Steam Heat.

The heat derived from the condensation of the steam may be distributed for warming buildings in many ways. (1) By means of a battery of pipes over which air is blown by a fan or blower, the heated air being conducted to the various rooms to be heated. (2) By a reducing valve fixed on the end of the pipe, reducing the steam pressure to a value at which it can be used in the radiators. (3) Tubular water heaters used as a water boiler, from which water circulates by ordinary low pressure gravity circulation. (4) By employing a coil fixed in a tank as a low pressure steam boiler, the pipes being led as in ordinary low pressure steam heating to the various rooms.

It is very rarely that high pressure steam can be advantageously employed for the direct heating of pipes fixed in the rooms, as the temperature of the steam is too great. This method is only employed for rough heating, such as factories, warehouses, etc.

Determination of Pipe Sizes.

The determination of the sizes of steam pipes necessary for carrying a required quantity of steam may be made by means of Figs. 133 and 134, which give the drop in pressure at various different pressures when certain quantities of steam are delivered through pipes of different sizes. It is well not to allow of too great a drop in pressure between the two ends of the pipe. If the main boilers are being worked at, say, 80 pounds no greater total drop of pressure should in ordinary cases be counted on than, say, 15 pounds per square inch. An even smaller total drop is preferable in the interests of uniformity.

Careful calculation should, of course, be made of the number of thermal units required, estimated on a liberal basis. These thermal units may be reduced to pounds of steam. Whatever units are adopted the necessary sizes of pipe can be at once determined. For instance, assume the pipe is 580 feet long, and a drop in pressure 20 pounds per square inch is allowed for, the initial pressure being 80 pounds per square inch.

In this case a drop in pressure of 20/5.8 = 3.45 pounds per hundred feet run, may be calculated on. Assume the total number of thermal units running on the far end is 800,000 B.T.U. per hour. The figure shows that the size of pipe required for this delivery is 1½ inches. Naturally, the higher the pressure maintained in the boiler the smaller the pipe which may be employed.

When heating alone is the service for which the steam is required it is much more advantageous to heat water near the point where steam is generated, and transmit the hot water by means of pumps or otherwise along the pipes. This is dealt with in another chapter.

In cases where large quantities of exhaust steam are given off by engines, especially electric light and power engines, the exhaust steam can frequently be utilised to great advantage in this manner, by heating the water and pumping the hot water to the distant points where heat is required.

Steam is frequently required for other purposes than merely heating, such as cooking, laundry work, disinfecting, etc. In these cases it is frequently *necessary* to carry steam instead of hot water to the distant point.

In calculating the sizes of pipes for heating, it is necessary to make a liberal allowance for the extra quantity of heat which is sometimes required by the necessity of heating the buildings rapidly in the morning. In many modern institutions it is the practice to maintain the heat on day and night, and in these cases smaller pipes can frequently be applied. Even, however, when the heat is turned off it is not infrequently the custom to leave the steam on the mains, in order to avoid the contraction and expansion on the pipes, which sometimes leads to leakages. This, of course, is a very extravagant practice. Its only justification is the inferiority of the expansion joints which cannot be relied on to prevent leakage.

For general high pressure distribution 80 pounds per square inch is found a convenient pressure. Air heating systems from 10 to 20 pounds per square inch is generally adopted.

Double Ranges of Pipes.

In some buildings where a continuity of heat is especially important, such as hospitals, etc., a double range of steam pipes is sometimes supplied, in order that if any repairs are needed to one of them, the other can be brought into operation. This arrangement, however, is very expensive, both in first cost and upkeep. Sometimes two different sizes of pipes are provided of which both together are necessary when the demand for heat is at its maximum, the larger size pipe at moderate temperatures, while the smallest pipe is alone used when the demand is at its lowest. This arrangement also enables the heating to be continuous and provides against a stoppage due to an accident or repair to either steam pipe. If an accident happens to the principal pipe, the heat supply must be reduced, if only the smallest pipe is available for supplying it.

The loss of heat from two pipes is always greater than the loss of heat from one pipe carrying as much as the joint capacity of the two. The higher the pressure of steam the smaller the pipe may be employed to carry the given quantity, and the less therefore the loss of heat. But since the temperature of high pressure steam is higher than that of steam at a lower temperature the loss of heat per square foot is greater. For the higher pressure, therefore, the advantage of increasing the pressure is diminished, though *with a good covering* the total loss of heat is less for a smaller pipe carrying a higher pressure than with a larger pipe carrying a lower pressure. This is not the case, however, with uncovered pipes.

Expansion Joints.

Great care must be taken in designing a range of steam pipes that free expansion and contraction are in all cases possible. The best kind of expansion joints to use are undoubtedly annealed bent copper pipes, but these are extremely expensive, and also are apt to become brittle and crack from prolonged use. Sliding sleeve type of expansion joint is frequently used on account of the smaller first cost. It is very difficult to obtain a sliding type of expansion joint which will not drip water continuously unless it receives constant attention.

At one end of each straight run of pipe a strong bracket is bolted to the pipe itself, or to the body of the expansion joint, and a bolted connection is built into the supporting wall. This then becomes a fixed point and the pipe is allowed to expand towards the expansion joint from the other end of the run. It is generally considered to be the best practice to allow a pipe to expand down hill, though it is difficult to find a rational reason for this practice. It is generally considered better to fix a pipe at one end rather than to allow it to expand in both directions, though in very long lengths of pipes between two water traps, some engineers prefer to bolt the pipe at its mid-point and allow it to expand in both directions. By this means the actual movement of any particular point of the pipe is reduced. A long length of steam pipes will generally expand 1½″ in every hundred feet, and this amount of expansion must in all cases be allowed for.

The most efficient kind of covering should be used in all cases for these long ranges of steam pipes. (See Chapter XVI. on the efficiency of pipe coverings.)

Pipe Supports.

There are various methods of supporting the pipe, so as to allow free expansion, perhaps the most common being horizontally fixed rollers upon which the pipes roll as they expand. A better method is to suspend the pipes by means of slings from a roof some two or three feet above the pipes. It is not, however, always easy to obtain supports for these slings.

Ranges of steam pipes can sometimes be designed so that expansion joints are not necessary. Thus, if a long range of pipes is in the form of an L it is sometimes possible to avoid the necessity for the application of expansion joints, by keeping the angle a considerable distance from the wall and allowing it to expand towards the corner, but in this case special arrangements must be made for the lateral movement of the pipes either by elongating the rolls or by making a special design of hanger, or by bolting to the pipe a bracket carrying a circular horizontal plate under the pipe. A corresponding plate is attached to a bracket built into the wall, and spherical balls between the two plates carry the weight of the pipe, allowing a free lateral movement in both the longitudinal and transverse directions. In any case, whatever design is adopted, the greatest possible care must be given to the design, to see that in no possible circumstances that any pipe be prevented from expanding freely.

In designing a system of pipes also, regard must be had to the possibility of a burst in the underground channel. If this should happen and no provision has been made for it, it may mean loss of life. An automatically closing valve is therefore a great advantage in such circumstances. These valves close themselves when an abnormal rush of steam past them takes place. Before being used for steam, all such ranges of pipes should be tested by water pressure to double the working pressure for high pressure systems, and to four or five times the working pressure for low pressure systems.

Condensed Water Pipe.

The condensed water pipe must be arranged to fall continuously in the direction in which the water has to flow. In some cases it is convenient to make a pipe run in the opposite direction, but in all these cases it involves the permanent water locking of the pipe at the dip. If frost should ever at any time reach such a pipe, a fracture will inevitably be caused.

The steam traps employed must be such that the steam cannot possibly escape into the water pipe, as this would give rise to great trouble.

All condensed water pipes should in any case be vented or be so arranged that they cannot become air locked. In situations where the building in which the heat is used is at a lower level than the boilers, special arrangements have to be made for the return of condensed water. The most convenient is an automatically regulated float pump, which discharges the condensed water into an auxiliary tank at a higher level than the feed tank, so that the water will flow from the auxiliary tank back to the feed tank near the boilers by gravity. Other favoured arrangements are electric pumps or steam pumps. In case steam is employed special arrangements must be made to condense the steam used by the pump. In most cases this can be usefully employed without wasting it.

When steam or electric pumps are employed for this purpose it is frequently advantageous to let the condensed water drip into a tank which carries a ball float, the ball float being connected to the steam valve or the switch controlling the electric power driving the pump in such a way that either the speed at which it works is regulated by the amount of condensed water returning, or the power is cut off when the level has fallen to a predetermined point.

The deterioration of condensed water pipes is frequently a serious matter. Instances have been known where a large range of condensed water pipes has been rusted through in one or two seasons. Especially is this danger to be feared where the pipes are made of wrought iron.

It is particularly important in the case of these pipes not to make joints by means of copper rings, as they set up an electrical action, which results in the speedy destruction of the iron of which the pipes are composed. Cast iron is better than wrought iron for condensed water returns, because cast iron does not rust so easily.

Condensed water pipes are generally partly filled with air. They do not commonly run full. The maximum corrosion takes place at the level where water, air, and vapour are in contact.

CHAPTER XXX.

ON HEATING BY STEAM IN A CLOSED SYSTEM OF PIPES.

Comparison of Principle of Circulation in Steam and Water Systems.

THE general principle according to which heat is distributed from a furnace to heating surface of whatever kind by means of steam is similar to that employed in a water system, but the great differences between the properties of water and steam introduce such variations into the principle by which the circulation is effected that the two are for practical purposes quite distinct.

In both cases, though in a different sense, the circulation may be said to be effected by differences in density between the heat carrier and the medium by which the rejected heat is conveyed for regeneration to the boiler.

In the case of a water circulation the density of the hot fluid conveying heat to the surface is only slightly different from that of the fluid conveying the rejected heat back to the boiler for regeneration.

This circumstance enables circulation by hot water to be effected without any very great differences of static pressure at the same level. The operative principle is a slight difference of density between the hot and the cool columns of water.

With steam, however, the case is quite otherwise. The large difference in density between steam and condensed water introduces conditions into a steam system which are fundamentally different from those existing in a water circulation.

The absolute pressure in a steam system has little or no relation to the head or height of the fluid column. In a water system, the pressure is almost entirely determined by that height. In a steam system the water sinks back to the boiler, not, as in a water system, by reason of a *slight* excess of pressure due to the greater mean density of the falling column, but because condensed water is so much denser than steam that the water column is *altogether* unbalanced by the weight of the rising steam column. The heat is distributed not because of this difference, but because owing to the abstraction of heat from the steam at the radiator and the peculiar properties of steam the absolute pressure at the boiler end of any steam main is maintained in excess of that at the radiator end of the same pipe.

In a water system the circulating pressure is generated in the system of pipes, in a steam system always in the boiler.

This circulating pressure in a steam system is produced by the radiator, not because, as in the case of water, the abstraction of heat increases the density of the medium, but because it reduces its pressure. The increase of density produced when the steam is condensed is an accidental feature, and not that which produces the circulation of steam.

If it were not for this circumstance steam would not move in the pipe at all. It is a necessary condition of continued circulation in a steam system that this continuous drop of absolute pressure all along the steam pipe must exist. In a water system this is not so; the variation in absolute pressure all along the flow main has no relation to the flow of water.

The changes in diameter of the steam pipe are not generally sufficiently great to set up considerable difference of dynamic pressure owing to changes of velocity in the steam pipe, but a considerable drop of pressure is in all cases caused by the generation of the high velocity at the entrance to the steam pipe. This loss of head is commonly not recovered owing to the steam being gradually reduced in velocity, as all subsequent changes of velocity, such as they are, are commonly sudden ones. There are cases, however, where the pressure at the end of a pipe is slightly greater than at the points where the velocity is highest.

The pressure in inches of water at 200° required to generate different velocities in a steam pipe is shown in the table :—

Pressure.	Velocity ft. sec.	Pressure.	Velocity ft. sec.
¼″	46.1	1½″	112.7
½″	65.2	2″	130.5
¾″	79.8	4″	184.3
1″	92.2	12″	318.0

In a water system the amount of heat conveyed by a cubic foot of fluid medium is about 3,000 B.T.U. as a maximum.

In a steam system the same volume of fluid conveys not more than 1/80 of this. The velocity of the fluid necessary to convey the same quantity of heat in a steam system as in a water system is therefore, for the same size of pipe, perhaps 50 to 80 times as great, according to the conditions.

In a water system the movement of the hot and return media are effected by the same agency. It is the same force which moves the hot water to and the cold water from the radiator.

In a steam system the movement of the return water is produced by its density, but the movement of the steam by its drop in pressure.

Principle of Gravity Circulation.

In order to explain this matter it will be well to consider the simplest case of a low pressure steam boiler feeding any device from which heat is abstracted for any purpose.

Consider, for instance, the system shown on Fig. 135, in which a boiler supplies steam to a pipe connected to a coil within a tank, the lower part of the coil communicates with the bottom of the boiler in the manner shown on the diagram.

Effect of Air in a Steam Coil.

Before steam is raised this coil is of necessity full of air, when a steam pressure is raised in the boiler the first effect is to compress the air in the system of pipes to the same degree. This air is quite cold and a large pressure may be raised in the boiler without any heat whatever reaching the coil in the tank. When the coil is full of air the pressure of the air is identical with that of the steam in the boiler and no steam passes up the pipe because the pressure at the beginning of it is the same as at the end.

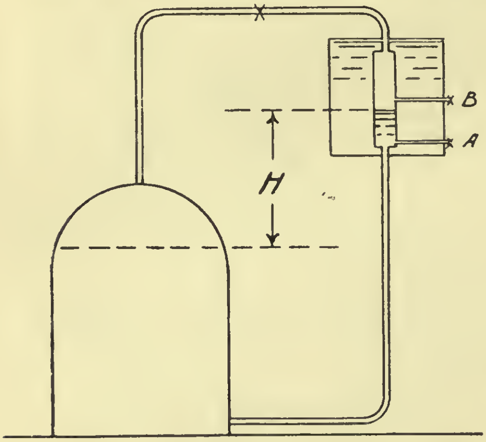


FIG. 135.

In order that the heat may reach the coil the first essential is to allow the air to escape from the coil through an air outlet.

In order to do this most effectively it might appear at first sight that the most suitable position for the air outlet would be at the bottom of the coil remote from the steam pipe at A, firstly, because steam is lighter than air at the same pressure in the proportion of 18 to 32. It would therefore be natural to suppose that the air would tend to sink and the steam to remain above the air, until all the air is pressed out by the advancing steam.

If the outlet is at the bottom one might suppose that the steam would follow the air down and the whole of the air be displaced before the steam reaches the outlet. But it must not be forgotten that when steam gets into the coil, water of condensation will be produced, which is even heavier than air, and therefore if the outlet is fixed too low it is certain that much water of condensation will be squirted out along with the air. A compromise is therefore generally adopted and the air outlet is fixed about halfway up the coil at B. In this position it does not get all the air out, but it does remove most of it.

Equilibrium of Pressures.

Now consider the balance of pressures between the interior of the boiler and the coil. Assume that all the air is displaced and that the steam pipe is sufficiently large to keep the coil always full of steam with only a minute and negligible loss of pressure between the boiler and the interior of the coil. In this case the pressure within the coil being approximately equal to that in the boiler, the water of condensation will at once fall back through the return pipe into the boiler by its own weight. There will be so little water of condensation produced, compared with the capacity of the return pipe, that the friction of this in the return pipe on its way to the boiler will be almost negligible. In any case, the distance by which the level of water in the return pipe will stand above the level of water in the boiler will be only that required to overcome the friction of the condensed water in the return pipe.

The more heat the coil abstracts from the steam and the smaller the steam pipe the greater will be the velocity in the steam pipe and the greater will be the loss of head in fluid friction within the steam pipe, and greater therefore the difference in steam pressure between the steam space of the boiler and the interior of the coil. The pressure in the boiler will be transmitted by the water in the return pipe to the free water level in the return pipe without any other reduction than is caused by the column of water between the free water level in the boiler and that in the coil. The pressure of steam on the free surface of water in the return will be reduced by the amount of fluid friction in the steam pipe. The consequence will obviously be that the water in the boiler will back up the return pipe and the system of forces in play will automatically cause it to stand at such a height in the coil above the water level in the boiler, that the actual reduction of steam pressure between the interior of the boiler and the interior of the coil is equal to the pressure due to a column of hot water of height H, increased by such a fractional height as is necessary to force the condensed water along the return pipe into the boiler.

Suppose, for purposes of explanation, that the whole of the air is removed from the coil by some means and that saturated steam at the boiler pressure is admitted, the coil being at a considerable height above the boiler.

It has been explained on page 48 that one of the peculiarities in the behaviour of steam is that if heat is abstracted from a mass of steam and its temperature thereby reduced, its pressure is at once reduced too, and that if the pressure is maintained and the temperature reduced, the whole of the steam collapses at once into water.

In this case then the presence of the cold walls of the coil will abstract heat from the steam and tend to reduce its temperature. The pressure will at once tend to fall and at once the balance of pressures in the pipe is disturbed. As the higher pressure is maintained in the boiler the steam commences to rush along the pipe at a high velocity in the effort to produce equilibrium.

In this movement some of the energy of the steam is absorbed in producing the velocity. Another part is converted into heat by friction within the pipe, and the usual natural process of the automatic solution of simultaneous equations commences.

The cold coil absorbs a certain amount of heat, depending on the exposed surface and the temperature of the steam. This condenses a corresponding amount of steam and causes a certain reduction of pressure. The drop in pressure reacts upon the temperature and reduces the amount of heat flowing through the coil. The drop in pressure also reacts on the velocity of steam within the pipe. As the pressure falls the velocity of steam becomes greater and greater until a balance is set up between the drop in pressure and the sum of the loss of head, due to acceleration and the frictional resistance of the pipe.

Similarly, the abstraction of steam from the boiler causes the water to boil more violently, and a consequent reduction in temperature and therefore in pressure in the boiler itself.

Eventually these variables adjust themselves in such a way that the water in the boiler gives off such an amount of steam as to supply the amount of steam flowing along the pipe, and at the same time maintaining such a pressure in the steam space as is necessary to set up the velocity in the stream of steam and overcome the friction of the pipe when steam blows through it at that velocity. At the same time the amount of heat flowing through the coil regulates itself to a corresponding degree and the pressure in the coil adjusts itself automatically, so as to maintain at the same time the temperature head, causing the flow of heat through the coil and the difference of pressure head causing the steam to flow through the pipe.

Equilibrium of Water in Return Pipe.

Consider now the equilibrium of the water. As fast as water is condensed from the steam it drops to the bottom of the coil and runs into the water pipe. The pressure on the free surface of this water is that in the interior of the coil, and the pressure at the other end of the return pipe is the steam pressure in the boiler, plus the head due to the column of water in the boiler.

The fluid actually in the boiler is not really solid water, but an emulsion of water and steam bubbles of smaller density than water. The effect of this is that the water in the coil does not stand as high *above the boiler level* as it otherwise would. The apparent water level in the boiler *rises* when the water commences to boil, since the same weight of water is in the boiler as before the emulsifying process commenced. The boiling of the water does not therefore affect the absolute water level in the coil.

Thus, the water in the return pipe and in the boiler forms a sort of irregularly shaped U-tube down which the water of condensation is constantly flowing. In order that this U-shaped column of water may balance itself, the difference in steam pressure on its two free surfaces must be exactly the same as the difference of pressure set up by the difference in the relative level between the two surfaces and differences of density in the fluid contained in the two limbs. In addition to this there will be a small difference of pressure introduced by the friction of the condensed water flowing through the U-tube towards the boiler. Ultimately, all these variables so adjust themselves automatically that a perfect balance is obtained in all these respects.

In order to make this important matter perfectly clear consider the following :—

Let p be the steam pressure in the boiler in inches of water column. This is then the pressure of steam on the surface of the water in the boiler.

Let $\triangle_{ps}$ be the reduction in steam pressure measured in inches of water column between the boiler and the coil, necessary to set up the flow of steam, Q pounds of steam per hour through the steam pipe and overcome the frictional resistance of the pipe.

The pressure of steam on the free water level in the return pipe is $p - \triangle_{ps}$.

H is the height in inches above the level of water in the boiler at which water would stand in the return pipe, if the return pipe were so large that there was practically no frictional resistance in it when conveying Q pounds of condensed water per hour, neglecting the effect of the emulsification of the water in the boiler. Then

$$\triangle_{ps} = H$$

If, however, the return pipe is so small that the passage of Q pounds of hot condensed water per hour through it would cause a loss of head of $\triangle_{pw}$, it is necessary that the height H of the column of water should be increased by $\triangle_{pw}$ in order not only to balance the loss of steam pressure $\triangle_{ps}$, but also to force water Q pounds per hour into the boiler.

The level of water in the condensed pipe then so adjusts itself as to stand above the water level in the boiler by

$$\triangle_{ps} + \triangle_{pw}$$

This tendency for the boiler water to back up in the condensed pipe is one that sometimes causes a good deal of trouble and needs a good deal of experience to deal with.

The extent to which the water will back up can only be calculated if (1) the total condensation of steam in the coil is accurately known, and (2) if the total loss of head due to the passage of this same quantity of steam through the steam pipe can be accurately calculated, which is, of course, not generally possible, and (3) if the resistance of the return pipe is accurately known.

It will be seen, too, that if the coil is so low above the level of water in the boiler that the effect of the backing up of water into the coil is to reduce the area of the coil which is exposed to steam, the amount of heat communicated from the water-filled space in the coil through the substance of the coil is so much less than that conveyed through a corresponding surface of steam-filled space as to be almost negligible in comparison.

The amount of heat which is transmitted through the coil then depends almost entirely on the proportion of the surface of the coil which is waterlogged. When the coil is at a small height above the water in the boiler, the transmission can be controlled by increasing the resistance of the steam pipe by partly closing the valve and so causing more water to back up into the coil. The same effect can be secured when the coil is high above the boiler by providing an auxiliary water tank so connected that a rise in steam pressure in the coil forces the water out of the coil into the tank, an overflow pipe in the latter connects with the boiler.

If a radiator is substituted for the coil the relative reduction in transmission is not so great. The transmission between steam and water through the substance of the coil is very much greater than that between steam and air. Also the transmission between water and water in a coil is greater than between water and air in a radiator. But the *proportionate* difference is not so great in the case of water as in the case of steam, so that the proportionate reduction in transmission, owing to a *coil* being waterlogged, is greater than that where a radiator is waterlogged.

Open Return System.

There is quite an analogous process whereby the quantity of heat delivered from a radiator or coil may be controlled. Instead of causing the water of condensation to back up into the coil, a somewhat similar effect may be produced by causing *air* to back up into the coil. The outlet is kept freely open to the air, with or without a steam trap. In this system the total pressure of the mixed air and vapour is always exactly the same as that of the atmosphere, say, 14.7 lbs. per square inch. The water of condensation passes away, preferably, of course, back to the boiler, the system being so arranged that it is liberated at such a level that it will force its way back into the boiler by its own gravity.

In any case, if the water of condensation is released into the open air, if there is not enough steam passing into the radiator or coil to "blow through," so as to keep the whole coil filled with steam at a pressure of at least one atmosphere, the total conduction of heat through the material of the coil will depend on the relative amounts of steam and air in it. This is in effect the French system of open returns.

If there is in the radiator enough air to produce alone a pressure of, say, 13 pounds per square inch, the pressure of the vapour alone is 1.7 lbs.

per square inch. The temperature of vapour at that pressure, as seen from the diagram, Fig. 3*b*, will be about 120° F. This will be approximately the temperature of the radiator.

The difficulty of working this system uniformly consists in the maintenance of an exactly constant pressure in the boiler. The temperature depends on the exact quantity of steam which is forced through the valve. A very small increase in pressure in the boiler may cause a large increase in temperature of the radiator. If the pressure is very slightly higher than that required to force into the radiator the maximum quantity of steam that the radiator can condense, the steam will flow through and may escape into the open air.

Vacuum System.

If, on the other hand, the whole of the system is closed and made air-tight after all the air originally in the radiators and pipes has been displaced, then if the pressure in the boiler is allowed to drop, it may drop below the atmospheric pressure, and the circulation of steam still continues to the radiators and coils, in spite of the fact that the pressure is below the atmospheric. It has been pointed out that the boiling point of water is only 212° F., when the mean pressure is 14.7 lbs. per square inch. When water is heated in a closed system absolutely cut off from the atmosphere the atmospheric pressure has no influence on the boiling point. The circulation will proceed on exactly the same principle below as well as above the atmosphere. The atmospheric pressure has only an influence when there are leakages.

This is one method by which, in theory at any rate, the temperature within a steam system may be reduced below 212°. Many attempts have been made to work low temperature systems on this principle. They have not, however, met with very considerable success, because of the difficulty of absolutely excluding the air from the interior of the pipes.

If a leakage of air occurs into a system which is working at low power at a pressure below the atmospheric, the effect will be that the entering air will drive back the whole of the low pressure steam within the pipes from the parts surrounding the leakage to other parts of the system. All the parts into which air leaks will then be left quite cold. The cold surface will gradually spread until the pressure within the system is the same as the atmospheric. The steam will be forced back into such a portion of the total surface nearest to the boiler as will condense at 212° all the steam which is being generated at atmospheric pressure.

It is, therefore, necessary for a satisfactory distribution of heat in this system that there should be no leakages whatever, even of the smallest kind, and this is a condition which it is very difficult to maintain. A vacuum pump or air exhauster may be provided to prevent the accumulation of accidental leakages of air into the system. To work such a pump requires a constant supply of power which is an undesirable complication.

Another method of expelling accumulations of air is periodically to raise the boiler pressure and expel the air through non-return valves, afterwards allowing the pressure to drop again below the atmospheric level.

There are, therefore, three systems of low pressure steam heating which in their principles are fundamentally distinct. These are :—

(1) The closed circulation with air inlet valves, the pressure in which may be above, but can never be below the atmospheric.

(2) The system of open returns in which, although the pressure can never be below or considerably above the atmospheric, yet the fluid within the radiator itself may be a more or less intimate mixture of steam and air.

(3) The hermetical sealed system, in which the pressure may be either above or below atmospheric in which the object is absolutely to exclude all air, especially when the pressure falls below the atmospheric. Failing this the air must be extracted by mechanical means.

The considerations which govern the sizes of pipe in each of these cases are distinct.

The Design of a Steam Heating System.

The object aimed at in designing a system of steam heating is as in all other systems of heating that the boiler pipes and radiators shall be of just such a size that they will supply to each of the points and coils as much heat as is required at each and no more. The conditions, however, are determined by a number of practical considerations which are different in other systems of heating.

Boiler.

The general principles governing the size of the boiler have been explained in Chapter XIX. For an ordinary low pressure steam installation the amount of heating surface in the boiler may be calculated from the formula

$$S = \frac{1.1\ W}{w}$$

where W is the total heat required from the boiler, including the loss of heat from the pipes, and w is the amount of heat which may be relied upon to be transmitted from the fuel through the one average square foot of boiler heating surface.

If the fire is glowing and in direct contact with the primary heating surface a value as high as 7,500 B.T.U. per square foot, or even more, might be taken as the heat transmission. But such conditions cannot be maintained for any considerable length of time, and further, much of the heating surface in a modern boiler is not direct or primary heating surface, but is indirect, not receiving direct radiation from the glowing fuel.

In order, therefore, that satisfactory results may be obtained, it is not desirable to reckon a boiler at a higher transmission than 3,000 B.T.U. per square foot, the whole of the heating surface being calculated, though up to 3,500 B.T.U. per square foot is sometimes allowed.

The amount of fuel which will be burnt per hour in a general case may be then calculated on the assumption that 60% of the total heat in the fuel will be transmitted to the boiler, that is to say, the general formula for the weight in pounds of the fuel to be burnt per hour in such an installation will be

$$\frac{100 \quad w\,S}{60 \quad F}$$

where F is the calorific value of the fuel.

Unit of Steam Pressure Adopted in this Book.

In this work it has been thought desirable to adopt the inch of water column at a density of 60 lbs. per cubic foot as the unit of pressure for low pressure steam heating. All calculations are made with that unit. 1″ water column thus = 5 lbs. per square foot and 28.8″ of water column = 1 lb. per square inch.

Pressure Required in the Boiler.

The general pressure for which heating installations may be calculated depends to some extent on the maximum length of the installation. In no case should a low pressure heating installation be calculated for a greater pressure that 3 pounds per square inch, at which pressure the pipe sizes should be only sufficiently large to ensure the maintenance at all radiators of a steam pressure not exceeding ½″ of water column for the best installations, though up to 12″ of water column at the radiator valve is countenanced.

The following table is given by Rietschel as the desirable steam pressure corresponding to various lengths of installations :—

Installation up to 300 feet long, pressure 20″ water column.
,, 660 ,, 40″ ,,
,, 990 ,, 60″ ,,
,, 1600 ,, 80″ ,,

Size of Radiators.

For ordinary steam heating it is customary to calculate on a transmission of 250 B.T.U. per square foot per hour. The size of radiator for any given duty is therefore $\frac{W}{250}$ square feet. The heat given off from any pipe may be calculated as follows :—

$$W = \frac{D}{12} \pi \; l \; K \; (t - t_a)$$

where K has the value given in the table, or 2.46 for medium size of pipes. Since t is commonly 212° and t_a 60°

$$(t - t_a) = 152 \text{ and } \frac{\pi \; K \; (t - t_a)}{12} = 98$$

Hence the formula for the loss of heat from a pipe becomes

$$W = 98 \; D \; l \text{ B.T.U. per hour}$$

where D is the outside diameter of the pipe.

The values of this formula for different sizes of pipes are given in the table.

For an ordinary well coated pipe, one may reckon

$$W = D \; l \times 30 \text{ B.T.U. per hour.}$$

Pressure Required Within a Radiator.

It will be understood that so far as the heat emission is concerned the pressure maintained within a radiator is not of very great importance.

The disadvantages attending a high temperature of heating surface have been already alluded to in the chapter on the emission of heat by radiators.

An installation in which the pressure in the radiators is allowed to exceed 10 pounds per square inch as outside limits is very rare, the more general pressures within a radiator being from zero to 2 pounds per square inch above atmospheric.

The temperature of steam corresponding to atmospheric pressure is, of course, 212°, while that corresponding to a pressure of 10 pounds per square inch is 240°.

According to the theory of a radiator, the rate of emission per square foot is approximately proportional to the difference between the temperature of the steam inside and of the air outside the radiator.

Comparing, therefore, the emission of heat from a radiator provided with steam at atmospheric pressure with that from one supplied with pressure at 10 pounds per square inch, their rates of heat emission per square foot vary as (212 — 60) : (240 — 60), that is, as 152 : 180—a difference of nearly 12%.

It will thus be seen that in any scheme of steam heating, in which the steam pressure is equal to the atmospheric, any reasonable rise in pressure can only result in the emission of the radiators being increased by about 12%, and this is not nearly a sufficient variation to allow for the difference in requirements corresponding to variations in external temperature.

It is only when the pipes are so designed that a low and fairly constant pressure is maintained at the valve, and a partial steam pressure in the inside of the radiator, that any considerable reduction or control over the temperature can be secured.

It has come, therefore, to be an accepted maxim in this country that it is impossible to control the temperature of steam heated radiating surface otherwise than by shutting off the radiator altogether.

Temperature of Heating Surface.

Steam at atmospheric pressure has necessarily a temperature of 212°. This alone is considerably higher than is desirable. The only methods of reducing the temperature in the pipes and radiators below 212° have been already explained.

It is clearly not desirable that a considerably greater total pressure than the atmospheric should exist in any radiator.

The problem, therefore, reduced to its simplest terms is that of providing at the valve leading to the radiator such a pressure of steam as will force through the valve when fully open a quantity of steam sufficient to maintain the entire surface of the radiators at the desired temperature at a pressure not exceeding the atmospheric.

Radiators are made with heating surfaces varying from a few square feet to several hundreds. Valves intended for this range of radiators are generally made in three or perhaps four sizes. Commonly no valve smaller than ½″ is used for steam heating, though occasionally ⅜″ are employed. 1¼″ is about the largest, though occasionally 1½″ are employed for very big coils.

It will thus be seen that there are only four sizes of valves commonly employed, ½″, ¾″, 1″, 1¼″, and one or other of these valves must be suitable for many hundreds of different sizes of radiators.

Sizes of Connections to Radiators.

The size of the connection to a radiator should be determined by two considerations, (1) the steam pressure in inches of water allowed at the entrance to the radiator, (2) the desired conditions within the radiator.

In order that it may be possible to control the amount of heat given off by a radiator by means of the valve, it is necessary that the radiator should condense all the steam which will be forced into it when the valve is wide open, and leave no surplus of pressure within the radiator. The pressure within the radiator should not be greater than atmospheric. In this case when the valve is partly closed enough steam cannot get into the radiator to expel all the air. The radiator will only partly be filled with steam and partly with air. If in these conditions steam is admitted at the top the result will be that the top of the radiator will be filled with steam and will be hot, and the bottom with air and will be cold.

If, however, the steam is admitted at the bottom and arrangements are made for mixing the air and the steam, the radiator may have throughout a moderate temperature. This will be liable to vary considerably with variations in the boiler pressure. Although in this way a certain degree of control may be obtained by partly closing the valve of a steam heated radiator, it is never possible to be sure that a given valve opening corresponds to a given temperature, because the latter depends on fluctuations of pressure in the boiler.

However, on the assumption that the maximum size of opening is to be calculated just so large that at the designed pressure the whole of the steam admitted by the valve is condensed within the radiator, there being no surplus pressure above atmospheric, it is necessary to note that if the velocity through the valve is too high (above 800 to 900 inches per second at the outside) the steam in passing through the small opening will cause a hissing noise, which is very unpleasant. A high velocity is produced by too great a difference of pressure on the two sides of the valve. The best calculated installations have a pressure of about ½″ to 1″ of water at the valve. The smaller the pressure at this point the better is the design.

The size of valve is, of course, determined according to the pressure available at the valve for forcing steam into the radiator. The greater this pressure is the smaller the valve may be. It is a common mistake in steam installations to make the steam pipe leading to the radiator too large, so that the pressure at the valve is too high. If any radiator is fixed at a considerable distance from the boiler and at a comparatively low level, it is not possible to heat it effectively with steam, unless the boiler pressure is very low or the pipes very large, otherwise the condensed water will back up into the radiator and prevent steam from reaching it.

All these facts must be taken into consideration in designing a steam heating installation.

Assuming the radiator is in the form of a continuous length of pipe it is easy to calculate what is the theoretical size of pipe which will suffice to deliver just so much steam into the coil that it is all condensed before the end of the coil is reached.

From the formula

$$p_2 - p_1 = \frac{0.626\ l}{(100\ d)^5\ \gamma}\ (3\ Q^2 + Q^1\ (3\ Q + Q^1)\ )$$

where Q' is the amount of steam at the beginning and Q at the end of a pipe, C.G.S. units, putting $Q = O$, we have as follows :—

$$p_2 - p_1 = \frac{0.626\ l\ Q_1^2}{(100\ d)^5\ \gamma}$$

This calculation gives the following table, due to Rietschel, for the necessary diameter of radiator connection, corresponding to various pressures at the valve :—

Diam. of pipe.		B.T.U. emitted by radiator or coil.											
		4,000	8,000	12,000	16,000	20,000	24,000	28,000	32,000	36,000	40,000	50,000	60,000
½″	Steam pressure at valve in inches water column.	.394	3.07	10.4	24.7	48.1	83.1	132	197	281	385	752	1,300
¾″		.0394	.394	1.34	3.19	6.22	10.75	17.1	25.5	36.2	49.8	97.3	168
1″		—	.118	.35	.827	1.615	2.80	4.41	6.58	9.37	12.8	25.2	43.4
1¼″		—	—	.039	.158	.276	.472	.748	1.10	1.575	2.16	4.22	7.32
1½″		—	—	—	.039	.118	.197	.315	.473	.710	.945	1.85	3.23
2″		—	—	—	—	—	.039	.079	.118	.197	.236	.473	.827

The following table gives the size of valve necessary in ordinary cases :—

FOR LOW STEAM PRESSURES AT VALVE (ABOUT ½″).

Size of valve, in. ...	½	¾	1	1¼	1½	2
B.T.U. from radiator	4,000	8,000	12,000	24,000	32,000	50,000
Approximate surface of radiator ...	16	32	48	96	130	200

FOR HIGH STEAM PRESSURES AT VALVE (UP TO 12″).

Size of valve, in. ...	½	¾	1	1¼	1½	2
B.T.U. from radiator ...	12,000	24,000	36,000	50,000	70,000	100,000
Approximate surface of radiator	48	96	140	200	280	400

As mentioned above the velocity of steam passing through a radiator valve, in order that the noise above alluded to may not be objectionable, should not exceed about 800 to 900 inches per second.

The heat emission per square foot at ordinary temperature is about 250 B.T.U. per hour per square foot.

The quantity of steam required per square foot per hour is therefore

$$\frac{250}{965} = 0.26 \text{ lb.} = 6.85 \text{ cub. ft.} = 12{,}800 \text{ cub. inches,}$$

or 3.3 cub. inches per square foot per second.

Hence, where a is the area of the valve per square foot of radiator

$$a \times 800 = 3.3$$

or

$$a = \frac{800}{3.3} = .00413$$

It therefore appears that the following is the approximate range of *minimum* sizes of valves required for radiators at the given sizes.

Since .0043 square inches corresponds to a total emission of 250 B.T.U. we have

$$\text{B.T.U.} = a \times \frac{250}{.00413}$$

or approximately

B.T.U. $= a \times 60{,}000$, where $a =$ area of valve in square inches.

		B.T.U.	Approximate size of radiator. Square feet.
½″	up to	11,800	50
¾″	,,	27,000	100
1″	,,	47,000	190
1¼″	,,	74,000	300
1½″	,,	106,000	420

The above are the minimum sizes in order that a velocity of 800 inches per second may not be exceeded. It is generally desirable to provide one size larger than these in order still further to reduce the inlet velocity.

In all cases where a large number of radiators have to be fed from a given system of mains, all the valves should be of double regulation type, so that the resistance of the pipe leading to each radiator may be controlled to the desired value. Any increase of resistance of this pipe, such as is brought about by partial closing of the valve, will then result in an insufficient supply of steam to keep the radiator quite full, and a certain degree of control may be thereby secured.

The degree of this control varies according to the connection at the other end of the radiator, and according to the height above the boiler.

Design of the Pipe System.

The objects aimed at in designing a system of pipes are three—

(1) To provide and maintain at the entrance to each valve in the system such a pressure as will silently force into each radiator the required quantity of steam.

(2) The system of pipes to be as inexpensive as possible and of such a size that the flow of steam along them shall cause no noise.

(3) To provide for the silent separation of all water of condensation, whether this is deposited in the pipes or radiator, and for its immediate return to the boiler.

The latter is a point of essential importance. A certain amount of steam is condensed in every inch of pipe along which the steam passes. It is clear, therefore, that the fluid travelling along any steam pipe must in fact be a mixture of steam and water. It can only be dry steam in cases when the steam is considerably superheated to such an extent that the superheat alone will provide for the loss of heat from the pipes. The latter is usually difficult or impossible to effect.

One of the most important features in the design of a system of steam pipes is that there should be no spot in the entire installation where the condensed water can collect. If there is it will certainly cause trouble. The criterion of good design in this respect is that if water is put in the pipe at any point it will at once flow continuously down into the return pipe and so into the return pipe below the water level in the boiler.

In regard to the first object named above, the magnitude of the pressures in boiler and at the valve have been already discussed.

In so far, therefore, as the sizes of pipes are concerned the problem resolves itself into providing such a system of pipes that when the selected pressure is maintained in the boiler and when the required quantity of steam is flowing along each, the total drop in pressure between boiler and radiator valve shall be equal to that desired. The criterion of good design in this respect is that when the apparatus is working at full power at its desired pressure, the pressure at the entrance to each radiator shall be the same.

The importance of uniformity of pressure at all radiators is partly due to the considerations which have been alluded to already.

There is another very important reason which is as follows :—

If any considerable pressure is admitted into any radiator not only will little advantage be secured from it in point of heat transmission from the radiator itself, but the pressure will be transmitted from the radiator and through the return pipe into the main return.

If then there are any other radiators in the system which have a lower pressure than the radiator in question, the steam from the first radiator will back up the return pipe, and flow by way of the return into the other radiator. On its way it will, of course, meet with condensed water coming from other radiators in the system, and will not only cause a noise, but may stop the flow of steam into the second radiator by the orthodox channel.

It is principally for this reason that it is important in all steam jobs that the pressure in all radiators should be about the same.

If these calculations are correctly made the steam pressure will be transmitted to each radiator valve uniformly, but only at pressures at or above that for which the calculations are made. If the pressures fall below this value, there will no longer be equality of pressure at the respective valves.

When the pressure is raised above the calculated value all radiators in the installation will have a higher pressure transmitted to them than the calculated value. For such increase in pressure as is here contemplated no considerable increase will be produced in the condensation of steam at the radiator. The pressure even at distant radiators will be increased at almost the same rate as the pressure in the boiler. The pressure at the near radiators will increase at approximately the same rate.

If, on the other hand, the boiler pressure is below the calculated value, the pressure in the distant radiators will fall more rapidly than the pressure in the near radiators. If the pressure is not sufficient to drive along the main pipe sufficient steam to provide for the condensation which will take place in the main pipe, then no steam pressure at all will reach the remote radiators and they will remain cold, although the radiators near the boiler may be almost as hot as when the pressure is higher. The reason for this will be understood from the following :—

Let Fig. 136 represent a steam heating installation in which there is one distant radiator and one near the boiler. The system is so designed that the pressure at the entrance to the valve shall in each case be 1″ of water when the boiler pressure is, say 36″, represented by O A.

From the previous discussion of the flow of steam in pipes it will be clear that the main pipe leading to each radiator has to carry not only the steam required by that radiator, but also the steam condensed in the pipe itself.

The amount of steam condensed in the short branch near the boiler is so small as to be almost negligible. Suppose, for instance, that the steam required by each radiator is 20 lbs. per hour, while the pipe leading to the distant one condenses 10 lbs. per hour, and that leading to the near one 1 lb. per hour.

In these circumstances suppose the steam pressure at each valve is,

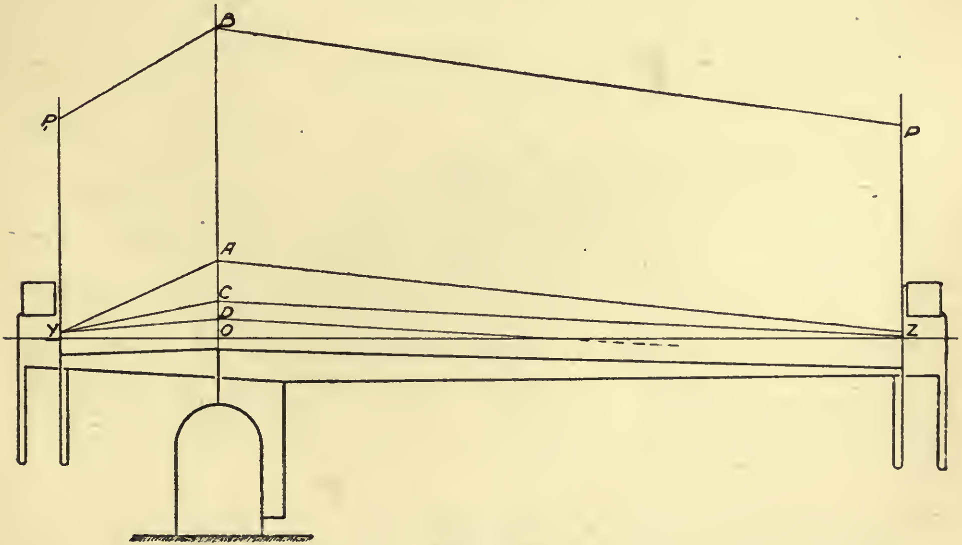


FIG. 136.

say, 1″ of water. The curves show the variation of pressure right along the main.

Assume that the pressure is raised to four times its original value, say 144″ of water or about 5¼ lb. per square inch, represented by O B. This would suffice almost to double the flow of steam along each main, but the radiators are not capable of condensing much more steam than at a lower pressure. Suppose the increase is 10%. The loss of head between boiler and the remote radiator would be about $35 \times (1.1)^2 = 42.5''$. The pressure there would be 101.5″, represented by Z P.

At the near radiator the pressure would be *Y P*, slightly higher.

Now let the boiler pressure drop to one quarter of its original value or 9″ of water column, represented by O C.

The quantity of steam passing along any pipe varies nearly as the square root of the difference in pressure at its two ends. Hence the amount of steam passing along each pipe will be reduced by about one half.

There will be 15 pounds passing along to the distant radiator and 10.5 pounds to the near one.

The quantity condensed by each pipe will hardly be affected at all. The long pipe will still condense 10 lbs., while the short one will condense 1 lb. There will thus be 5 pounds left for the distant radiator, but 9.5 pounds for the near one.

As the valve opening will be the same in the two cases, the pressure maintained at the distant valve will be

$$1'' \times \frac{5^2}{20^2} \text{ or } .0625''$$

while the pressure at the near valve will be

$$1'' \times \frac{9.5^2}{20^2} = 0.226''$$

or nearly four times that at the remote one.

When the pressure falls to 4″ of water, represented by O D, it is so small that the whole of the steam forced along the main is condensed in the main and none is left for the distant radiator. The lower curve will die away into the base line at a certain point near the remote radiator, and all radiators beyond that point must necessarily be left quite cold, since there is no pressure left to drive steam through them. The near radiator, on the other hand, would carry 7 pounds of steam per hour, of which 6 would be left for the radiator. The pressure at the valve would be

$$1 \times \frac{6^2}{20^2} = .09''$$

The cause of the difference is that the surface of the flow pipe which conveys the steam is always full of steam, and any reduction in pressure at the boiler only reduces the condensation in it to a very slight extent, so that when the pressure in the boiler at any time becomes so low that the flow pipe is not large enough to pass enough steam with this low pressure behind it, to keep the flow pipe and the distant radiators both hot together, then the flow pipe will get it all and the radiators none.

The near radiators are not dependent on the flow pipe to the same extent. The near radiator has the same preferential pull on the available steam as the flow pipe leading to the distant radiators.

It will be evident from the foregoing discussion that the better the coating which is employed on the steam main the more uniform will be the result at the distant radiators, for the smaller will be the proportion of the total heat which is lost on the way to them.

It is thus seen that it is impossible to design such a system of pipes that a uniform circulation will be secured at all pressures, and impossible to control the distribution of heat in proportion merely by regulating the intensity of the combustion. A uniform distribution of heat will only be secured at one particular pressure, for all other pressures the pipe sizes will be incorrect.

Determination of Pipe Sizes in Practice.

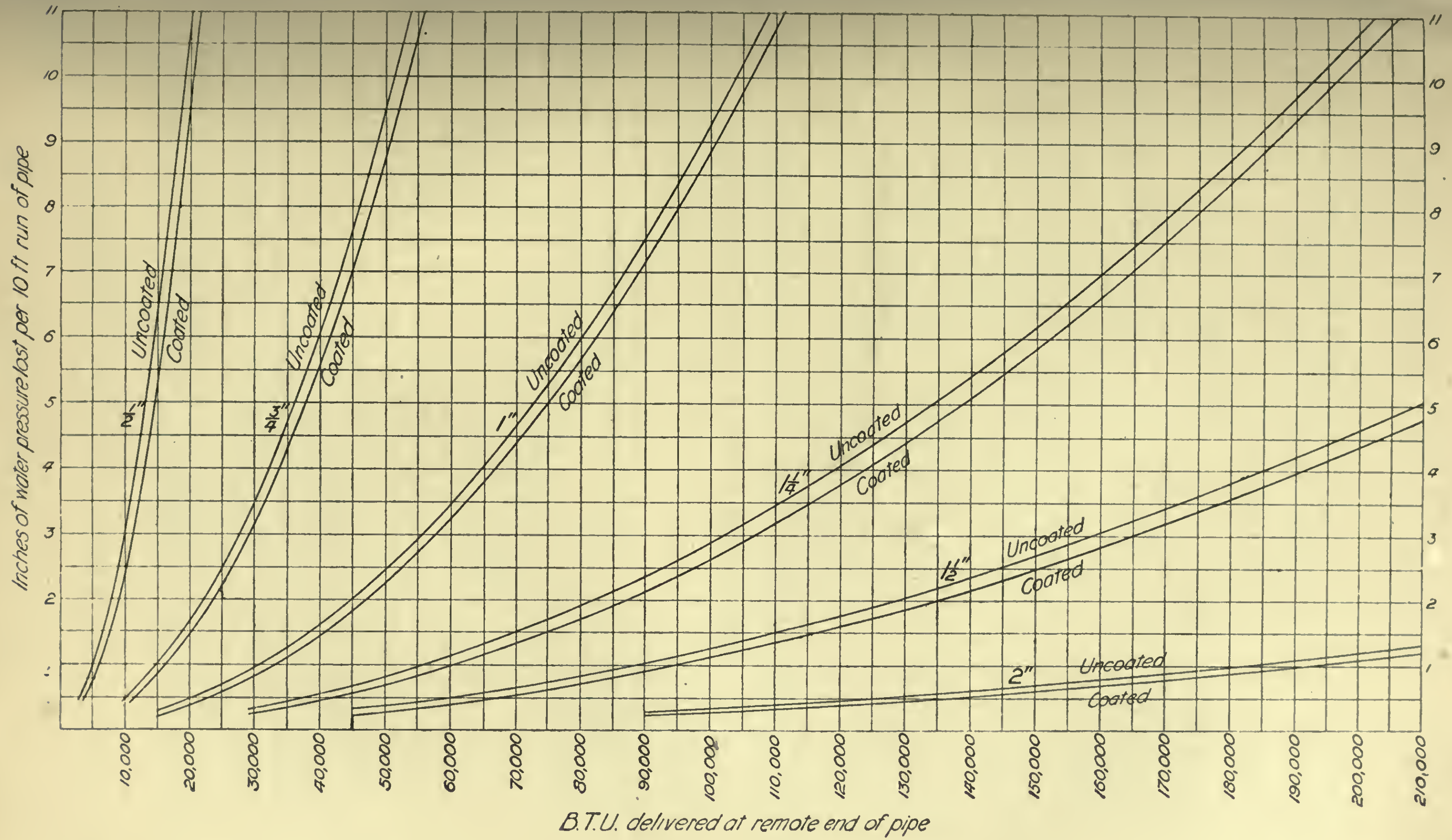
The positions of the pipes having been laid down on plan and the number of B.T.U. required at each point having also been figured, it is first necessary to make an approximate determination of the sizes in order that the final calculations may be accurately made, as in the case of the hot water installation.

For the preliminary calculation the sizes of pipes not being known, it is therefore impossible to determine directly the amount of heat lost from each pipe length.

The approximate determination may therefore be made by adding in succession up the total B.T.U. required from each radiator, and adding to the sum so obtained a figure varying generally from zero up to 30% of the total, which percentage represents an approximate guess at the total heat lost from the pipe. Thus, in a case where it is estimated that 30% of the total heat will be lost from the pipes, the main pipe leading from the boiler would be figured 30% in excess of the total obtained by adding up the B.T.U. from the radiators, half way along the main 15% might be added to the total, and so on.

The approximate sizes of the pipe may then be determined directly from the table.

It is customary to make the size of the return pipe as follows, according to the size of the steam pipe :—

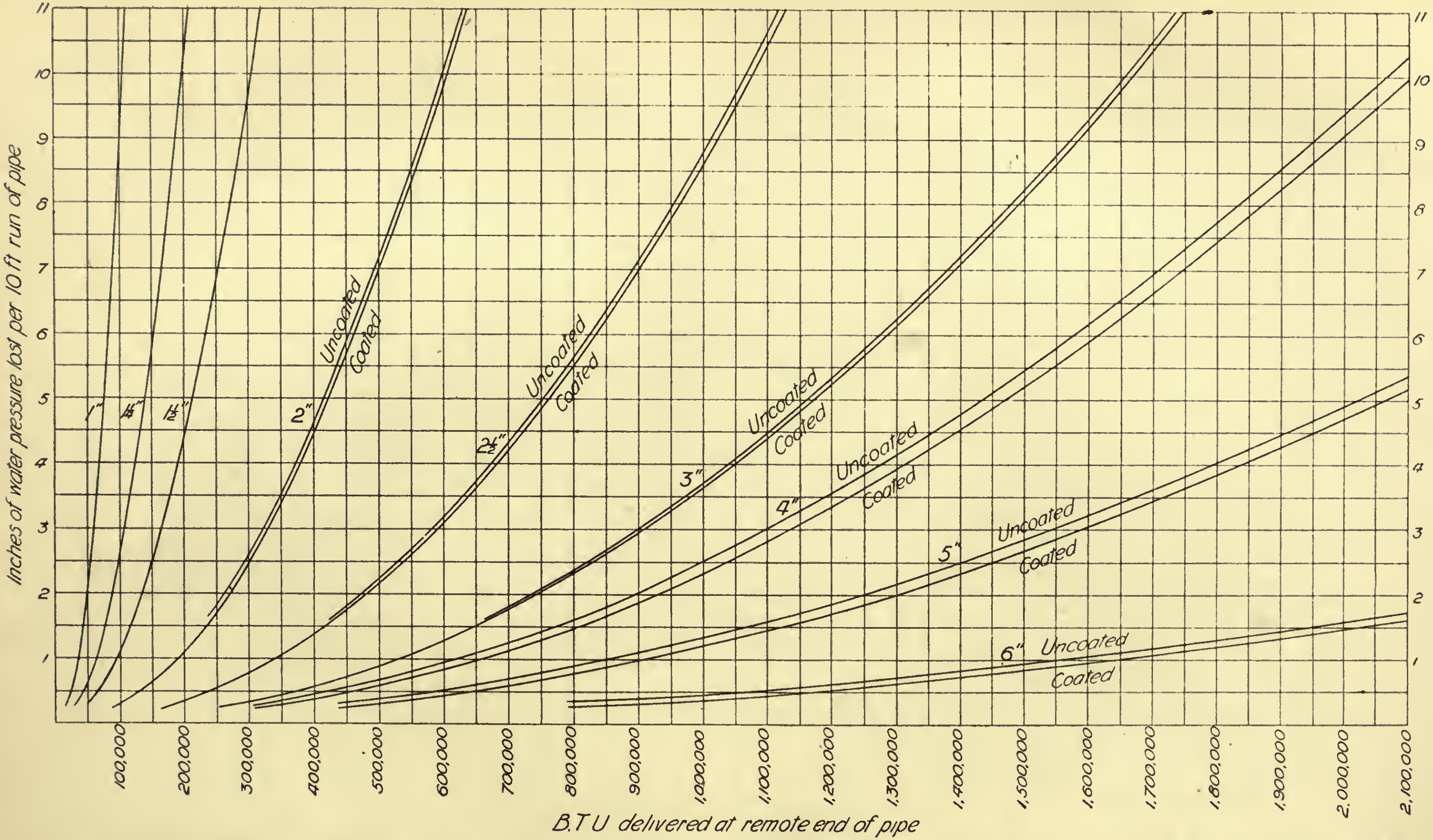


LOW PRESSURE STEAM HEATING. APPROXIMATE LOSS OF PRESSURE WHEN VARIOUS QUANTITIES OF STEAM HEAT ARE DELIVERED THROUGH COATED AND UNCOATED PIPES

1. Small Pipes

A.H. Barker, delt.

FIG. 137.



LOW PRESSURE STEAM HEATING. APPROXIMATE LOSS OF PRESSURE WHEN VARIOUS QUANTITIES OF STEAM HEAT ARE DELIVERED THROUGH COATED AND UNCOATED PIPES

II Large Pipes

A.H. Barker, delt.

FIG. 138.

Size of steam pipe.	Size of return.	Size of steam pipe.	Size of return.
½″	½″	2″	1¼″
¾″	½″	2½″	1¼″
1″	¾″	3″	1½″
1¼″	¾″	4″	2″
1½″	1″	5″	2″

With a very long return pipe it is sometimes thought advisable to increase the pipe one size. Generally the above sizes are found larger than necessary.

Accurate Determination of Pipe Sizes.

The provisional determination having been made, it is now necessary to correct these sizes by exact calculations as to the loss of head in each pipe. These calculations have for their object so to reduce the pressure generated in the boiler that by the time the steam reaches the radiator valve, it has been reduced in pressure to about 1″ or 2″ above the atmospheric.

The calculations are similar in general character, but much simpler than the corresponding calculations for a water system, inasmuch as the available head for each circulation has not to be determined. The available head is given by the excess of pressure of the predetermined boiler pressure over that desired at the entrance to the radiator.

The best practical method of making the calculation consists in making an isometric projection of the entire installation. This is, of course, not always necessary, but it will be generally found to be an economy in the long run to make an exact isometric projection before completing the remainder of the figures, especially having regard to the convenience of this drawing for the erector and workmen and the fact that its production compels individual attention to every part of the entire installation whether shown on the plan or not. A workman provided with such a drawing generally needs to ask no questions whatever and to waste no time in studying the meaning of plans.

On the office copy of this isometric projection are figured in red ink the B.T.U. required from each radiator. Each pipe length in the entire installation is numbered and measured. The loss of heat from it is calculated on the assumption that its diameter is according to the provisional determination previously made. This calculation is made according to the formula on page 578. Preferably the values are taken direct from the Table XV.*a*.

These quantities of heat are also figured on the plan in red pencil so that they can be removed and the correct values substituted at a later stage, if any alterations are found necessary. When each pipe in the entire installation has been so treated the exact calculations of the heat carried by each pipe must be made by adding up in succession all the quantities of heat on each pipe, the result being figured on the isometric projection.

This operation is carried out on the same principle as has been already explained for hot water calculations.

The losses of head produced when the given quantities of heat are conveyed by means of steam through pipes of the assumed sizes, are then calculated by the aid of the tables, and the results figured on special form, a specimen of the ruling of which is given. The total loss of head between the boiler and the most remote radiator is then added up on this form. If this loss of head is less than difference between the boiler pressure and the pressure desired at the valve of the radiator, it is an indication that the sizes of pipes provisionally selected have been too large, and one or more of the component lengths of pipe must be reduced in size in order to consume more of the available pressure.

In making these alterations, it is desirable to observe that the chief drop in pressure should take place in the main steam pipe leading from the boiler to the first junction. By this means a more uniform distribution of pressure over the entire installation will be secured, and also the flow of steam through such pipes will be more silent. A relatively small drop in pressure per foot run is desirable in the interests of silence in the *small* pipes near to the radiators, especially where such pipes are fixed in the room to be heated.

A steam installation in which a high velocity is maintained in such pipes is sometimes intolerably noisy on account of the high velocity of the steam rushing through the pipes. Also a small pipe gives little facility for water to separate itself from steam.

The effect of too small a diameter of pipe, even though it may be theoretically large enough for requirements, and even though it is properly laid for draining, is sometimes liable to produce violent cracking noises, especially when steam is first introduced at the commencement of the heating. For these reasons it is desirable that the loss of heat should take place chiefly in the main pipes, which are large and not likely to cause these troubles.

When the sizes of pipes leading to the most distant or index radiator have been determined and also the quantity of heat flowing along all mains, the pressures at each junction can be determined and figured on the plan, generally in blue ink.

The circuit next to the index circuit can then be calculated in exactly the same way; starting from the junction, the total loss of head between the junction and the radiator valve should be calculated to be in such a way that the pressure at all radiator valves is the same.

When calculations are carried out in this manner the pressure throughout the system will be uniform, for even although the exact values of ρ are not correct, yet the *proper proportion* will be observed in the resistance of the pipes, and errors in the absolute value can be compensated for by increasing or decreasing as may be necessary to a slight extent the pressure in the boiler.

SPECIMEN RULING.

Size of Pipe.			Heat emitted B.T.U.			Loss of Pressure in ins. water col.		Corrections.				
No. of Pipe Length.	Length. Feet.	Assumed Diameter.	From Radiator.	From Pipe.	Total.	Per 10 foot run.	Total.	Corrected Diameter.	Corrected Loss of Heat.	Loss of Pressure per 10ft.	Total Loss of Pressure.	Grand Total.

CHAPTER XXXI.

THE PLENUM SYSTEM.

THE system of heating in which heat is communicated to the apartment by means of air forced over a battery of heating surface into the apartment, is known as the "Plenum System."

A similar system in which the air is drawn into the building by means of a suction fan, connected with the extract outlet, is known as the "Vacuum System."

One chief difference in effect between the Plenum and the Vacuum systems is that in the former the level of the neutral zone is lower than in the latter.

As has been previously explained, the forcing of air into a chamber from outside under a slight pressure, tends to depress the neutral zone, whereas the withdrawal of air from the room by means of a suction fan tends to raise it.

The apparatus necessary for applying the Plenum system consists essentially of a fan or air propeller with a battery of heating surface supplied with heat by furnace, steam, hot water, or other source, and a system of air channels for delivering the warmed air to the various parts of the building to be heated, or for distributing it in a single room or hall.

Other subsidiary apparatus generally supplied with a well designed Plenum system are filters or air washers, humidifiers, and other apparatus for controlling the quality, quantity, or temperature of the air delivered into the building.

The advantages and disadvantages of the Plenum system have been the subject of prolonged and acrimonious discussion on the part of medical authorities, heating engineers, architects, and clients. No general agreement as to its merits and demerits has been arrived at.

There are persons who from experience of Plenum heated buildings condemn the system completely. Others arrive at the same conclusion through an imperfect comprehension of the theory of ventilation. Others believe that the system is not only good, but is the only possible system whereby definite and certain results can be obtained in the way of supplying fresh air to large buildings, without, at the same time, giving rise to considerable inconvenience by draughts, noise, and other undesirable effects.

It is beyond question that there is truth in both views. There can be no doubt that many buildings have been supplied with Plenum ventilation appliances of the most expensive description and designed with considerable care and skill, which, owing to some reason, plain or obscure, have proved not only expensive in upkeep, but unsatisfactory in result, when judged according to the sensations of comfort produced in the rooms.

Complaints Against the Plenum System.

Devitalisation.

The chief cause of complaint against the Plenum system is the devitalising effect which it seems to produce on the air. This effect is more noticeable by some persons than by others.

This question has been already discussed in Chapter VI. The difference in effect on persons of different organisations probably accounts for the wide difference of opinion on this subject. Some persons are totally unable to appreciate any ill effects, while to others the feeling of devitalisation is very distressing indeed.

Variations of Temperature in a Plenum Heated Room.

Another effect, which gives rise to complaint, is due to the fact that when heated air is delivered into a room, its relatively low specific gravity causes it to rise to the ceiling. The heat only reaches the floor level by gradual displacement, during which the air is considerably cooled down by contact with the cold walls and ceiling. The result is that there is a continual fall of temperature between the ceiling and the floor. This causes a complaint from persons in the room that their heads are hot and their feet are cold. A very cold layer of air is frequently found on the floor, which may be 5° or 10° below the mean temperature at head height.

The existence of this layer is probably due to the fact that the neutral zone is above the level of the floor. Cold air is, in consequence, drawn in through the cracks between the floor boards. This layer has a tendency to cling to the floor, owing to its density and the difficulty with which heat is conducted downwards in a fluid.

The effect is less marked in a radiator heated room, or in any room in which the heat is delivered at the floor level, because the circulation of air so caused draws this stratum of cold air into the circulation, as explained in Chapter VIII. Where the heat is delivered at a high level the circulation is largely confined to the region above that level.

This effect is greater the higher the temperature of the air so introduced and the smaller its volume. It is also intensified if the extract air is taken from the room at any height above the floor level.

Humidifying of Air.

A further complaint is due to the fact that many otherwise well designed Plenum systems are not properly supplied with humidifying apparatus, and that therefore the air is made dry and causes a sensation of "parchedness."

Cost of Upkeep.

Yet one other serious cause of complaint is the high cost of upkeep. This cost, however, is not so much due to the Plenum system itself as to the fact that any system in which large quantities of air are passed through a room duly warmed and then passed away to the outside air, must necessarily be expensive in upkeep. Every thousand cubic feet of air raised through a certain range of temperature must of necessity absorb a certain quantity of heat, which therefore means the combustion of a corresponding quantity of fuel. The cost of this fuel cannot be laid to the charge of the Plenum system as such, but to the necessity of complying with hygienic demands. Whether the latter are or are not justified is not a question which the heating engineer should be called upon to decide.

Any system in which the same quantity of air is heated to the same degree would cost the same in upkeep. Of course, where an excessive cost is caused by the inefficiency of the method used for transmitting the heat to the air, this excessive cost must be laid to the charge of the particular apparatus used, and not to the Plenum system itself.

In addition, however, to the cost of fuel, there is the considerable cost of power. The Plenum system is commonly carried out by means of centrifugal fans driving air at considerable pressure and velocity through comparatively small ducts. The necessary power for maintaining this current of air varies as the square of the velocity, and if the power is supplied electrically it is, of course, expensive.

This cost again is not essentially a defect of the Plenum system, but of the way in which it is carried out. When large ducts are employed the cost of power is, of course, very much reduced, although the cost of installation is increased. There are plenty of persons who will insist on a reduction of first cost and afterwards blame the engineer for the consequent increased cost of upkeep.

The cost of a fan which will deliver a certain quantity of air at a relatively low pressure is greater than the cost of a fan *of the same type* which will deliver the same quantity of air at a higher pressure and with a higher consumption of power.

But when a very low pressure is employed a propeller type of fan can be employed instead of a centrifugal one, which is not only very much cheaper in first cost, but absorbs very much less power at low pressures than does a fan of the centrifugal type.

Noise.

The higher the speed and the higher the pressure at which a Plenum system works, the greater tendency there is to the production of noises, due to the air rushing at a high speed through the flues. If flues of large size are designed they are apt to be extremely difficult to accommodate in an existing building without producing very unsightly and inconvenient results.

The case is, of course, to some extent different where the system has to be applied to a new building, when the engineer has the plans before the work is carried out. In this case the necessary cost of the flues is considerably reduced, and by consulting with the architect the difficulty of finding place for them can frequently be easily overcome.

Unhealthiness of the System.

A more serious complaint against the Plenum system is sometimes heard from certain medical authorities, especially in regard to such buildings as hospitals and infirmaries, viz., that it is unhealthy.

If it is so, the actual cause of it is not so much a matter for the Engineer to determine as for the medical authorities themselves. It certainly cannot be due to the mere passing of large quantities of air through the rooms. It may possibly be due to some such cause as the following :—

The tendency of the Plenum system is to maintain a pressure within the building supplied, higher than the corresponding outside pressure, and higher, of course, than the pressure would be either on the natural system or the Vacuum system. Since no walls are absolutely air-tight, and since the quantity of air passing through the substance of the walls is proportional to the difference in pressure on the two sides (see page 184) this excess in pressure within the room must result in the passage of an enhanced quantity of air through the walls.

This cannot fail to carry with it some, at any rate, of the small disease germs which may be produced within the ward. These are ultimately deposited within the substance of the wall, where it is conceivable that they may multiply if they find a suitable environment.

The walls on this system therefore tend to become more permeated with disease than on either the natural or Vacuum systems.

A further cause of unhealthiness of the Plenum system, if such can be said to be proved, is that as commonly designed, the fresh air flues and the apparatus employed are either difficult to keep clean or the cleaning is commonly neglected by the person in charge, and not sufficient supervision is given to such persons by the authorities.

It is a matter of the first importance to obtain satisfactory results from an installation of this kind that not only the flues, but the battery and all parts of the fan shall be kept as clean as possible.

It has been stated by medical authorities that an examination of the dust and accumulation of dirt in fresh air flues discloses no living organic germs. The theory is that the passage of large volumes of air over such accumulations, has the effect of killing all the living germs. Whether this is so or not, it is certainly contrary to modern notions of hygiene, that air to be breathed by human beings, especially by invalids, should pass over any dirt or dust, be it organic or inorganic.

For a ventilating installation of the best class, provision should be made for cleansing not only the inlet, but also the extract channels. With a stoppage of the positive motive force producing the ventilation a reversal of the direction of current is apt to take place, causing an inflow of cold air into the room, through the extract outlet. This, of course, brings back into the room from the extract channels, the foul deposits from the vitiated air.

In practice it is very rarely that extraction flues are cleaned. Either it is thought unnecessary to do so, or the amount of labour involved prevents it being properly carried out. This is indeed one of the chief drawbacks of the Plenum system, that however clean the inlet flues are kept, it is very general to find the outlet flues never cleaned at all, and they are therefore almost always in an indescribably filthy state. It is highly probable that many of the complaints as to the unhealthiness of the Plenum system proceed from this cause.

Cleaning of Flues.

The necessity for providing facilities for cleaning the fresh air ducts is very troublesome and expensive. For thorough cleaning of such ducts, it is necessary that they shall be large enough for a man to creep up them. It is also very desirable, but very expensive, to line such a duct with glazed tiles, which do not collect dirt, and to provide means whereby they can be thoroughly flushed out with water from a hose.

The greatest difficulty is, of course, to keep clean the vertical ducts leading from the horizontal duct in the basement to the various rooms. These vertical ducts are generally of too small a size for a man to get up them. The only means of cleansing them, therefore, consists in the application of a brush or of a stream of water. This is far from satisfactory. It is probably due to the consequent fouling of these ducts more than to any other single cause that the bad repute in which the Plenum system finds itself is due.

Cleaning of Heating Surface.

It is not easy to arrange a battery of heating surface within a reasonably small compass, in such a way that it can be easily cleaned, and at the same time that the air must be forced *through* the battery and cannot escape from the fan into the flues without passing through the battery at all. In order to effect this a more or less complicated system of baffle plates and doors must be arranged, and the whole arrangement generally occupies an inconvenient amount of space, and is therefore correspondingly expensive in builder's work and in other ways.

But it is an essential condition of an apparatus of the best type that not only the flues themselves, but that all parts of the heating surface shall be easily accessible for cleaning purposes, and it is not sufficient merely to provide means whereby the surface can be more or less reached with a brush.

If the battery fills up the whole width of the heating chamber it is impossible for an attendant to reach each side of the battery otherwise than by the provision of a separate passage from one side to the other. If, as usual, the battery consists of a large number of pipes or radiators, access must be provided to several points at the centre of the battery, otherwise it is impossible to clean it.

To produce the best results it is necessary to clean a battery at least once a week, with a more thorough cleaning once a month and at the end of the season.

General Remarks on Arrangement of Apparatus.

1.—Position of Fan and Cleansing Screen and Battery.

In cases where the cleansing screen is of the wet type, it is necessary that the air should be warmed or tempered before passing through the screen. If this is not done the screen cannot be used in frosty weather, for it will cause all the water to freeze, and may do some damage by bursting pipes, etc.

It is therefore necessary that on its entry to the chamber the air should pass through a portion of the heating surface, so as to raise its temperature to 40° or 45°.

It is open to question whether it is better for the air to pass next through the fan or through the cleansing screen. In any case the air must pass through the cleansing screen before it passes through the main portion of the heating surface. If this is not done, not only is dirt deposited in the heating surface causing smells, but also the warm air will take up too much water, and will be delivered into the rooms too moist.

If it passes first through the screen the subsequent heating by means of the battery diminishes the humidity of the air, even though it is saturated on entering the battery.

If the screen comes before the fan some moisture from it may be carried into the fan bearings, which may cause a deterioration of the fan. But this is not generally regarded as a very serious danger.

It is generally more convenient to arrange the screen before the fan, for constructional reasons. The chamber into which the fan delivers is commonly not so large as the fresh air chamber, and the screen is required to be always as large as it can conveniently be made.

It is in most cases desirable that the air shall pass through the fan before passing over the heating surface, though there is no general agreement on this point. One reason for this is that the operation of the fan in most cases makes a slight noise, and this noise is liable to be most audible in the rooms if the fan is in direct connection with the flues. The interposition of the battery breaks up sound waves, and they are therefore not nearly so audible. The same consideration would, in the absence of reasons to the contrary, lead to placing the screen on the building side of the fan.

Further, the attention required to the fan and motor is much more conveniently given in a comparatively cool space. If the fan is fixed on the far side of the battery, any attention necessary to the motor must be given in an unpleasantly hot atmosphere, and in consequence the fan and motor are liable to get less attention, besides which the space required for the fan and motor are more conveniently provided in the larger part of the heating chamber, which is commonly that on the outside of the fan.

Also the lubrication of both the fan and the motor at the higher temperature is more likely to give trouble, and in cases where the fan is worked direct—coupled to a motor (which arrangement is not recommended on account of the greater liability to noise), the higher the temperature surrounding the motor, the less satisfactorily it is likely to work.

On the other hand, if the fan is placed on the cold side of the battery, both it and the motor are more apt to get wet, though this is not so serious a trouble as the other.

Remarks on the Arrangement of the Fan.

The fan must be of such diameter that it can deliver the requisite quantity of air without causing anything in the nature of a roaring noise, which would be audible throughout the flues, whether or not the fan were placed on the near or far side of the battery. The extreme speed for silent running varies for different types of fans.

In general the extreme speed of a fan of the centrifugal type should not be more than about 50 feet per second, which will commonly produce a total water gauge of ⅝ inches.

This limit is placed to the speed, both on account of the noise, and because it represents a medium quantity of power required.

The arrangement of the fan should be such that its power can be reduced either by a reduction of current or by alteration of pulleys when the extreme quantity of ventilation is not required.

It is very desirable that the space between the fan and the battery shall be so arranged that the air delivered shall have time to distribute itself over the whole of the heating surface, and that there shall be no local currents, but a fairly uniform velocity over the whole of the surface.

It is very desirable that permanent means shall be supplied for estimating both the quantity of air and its pressure, between the two sides of the fan, and that "speedometers" shall be provided so as to enable the caretaker to work the plant in an efficient manner.

In selecting a fan the engineer should be informed as to its effect at different speeds, and he should be able to calculate what quantity of air will be driven into the flues, and how it shall be distributed at three or four different speeds at which the fan will be used.

The chapter on "fans" explains the rationale of this calculation.

The pressure against which the fan works will generally be from ⅝″ to ¾″ of water pressure in installations provided with a centrifugal fan.

When provided with a propeller type of fan, the pressure should not be more than 1/16″ or 1/8″ water gauge.

This, of course, requires much larger flues.

Air Screens or Filters.

These are of different types, and each different type has its own special resistance.

It is not possible to give general figures whereby the resistance caused by any given screen may be accurately determined. They can only be determined by experiment in each case, but a test on this point should never be omitted, that is, simultaneous measurements should be taken of the air pressure on the two sides of the screen, corresponding to different deliveries of air per hour.

A permanent manometer or water gauge should be provided, so that the caretaker can see at a glance what is the difference of pressure on the two sides.

This pressure will vary according to the state of cleanliness of the screen.

After being used for a certain time, the accumulation of matter abstracted from the incoming air causes a pronounced increase in the resistance of the screen, and the indication of the water gauge should be sufficient, after a little experience, to inform the caretaker when the screen needs to be cleaned.

The figures in Table XXXIII. are quoted on the authority of Rietschel for the difference in pressure of various kinds of dry cloth screens.

A screen should be always designed as large as it is possible to be made within reason. The larger the screen the better it does its work, and the less resistance it causes. A common proportion between the size of the screen and the amount of air passing through it, is to allow at least one square foot of screen for every 5,000 to 10,000 cubic feet of air passing through it per hour. This is a mean value.

Twice as much air as this is sometimes passed through a screen where difficulty is obtained in getting the necessary area.

On the other hand, not infrequently screens are twice as large as this, so that no general rule can be given.

The fresh air inlet and all parts of the channel should be constructed in such a way that they do not give rise to dust, that is, if the walls are cemented, they should not have a finish which is liable to crumble away to cause dust in the air settling chamber. The velocity of the air in this air settling chamber should not exceed 3 feet per second, although installations are frequently provided where the velocity is considerably greater than this.

It is not always possible to adhere strictly to rules on such points as this, for they depend on the local conditions.

It is the common practice in the best installations that the air is taken from a point some 8 or 10 feet above the ground in a protected space, which should be as free from dust as possible, and for installations of the highest class it is not infrequent that the air should enter through a playing fountain, so as to cleanse and moisten the air on its entry as far as possible.

From this main inlet the air is led to the settling chamber.

None of the flues employed for the passage of air should be allowed to be used for ordinary gangways for workmen or others, otherwise it would be impossible to keep them clean.

Heaters.

The method of calculating the size of the battery is given in Chapter XXI.

The battery should be arranged with special reference to facilities for cleaning.

In the best modern plant it is possible to reach all parts of the battery by hand. This, of course, means greatly extended battery space.

A common form of battery consists of 1″ pipes arranged in rows, having a space of 5/16″ to ½″ between the outsides.

This form of surface, however, is being gradually displaced by the cheaper sectional cast iron radiation, which is now made in convenient units for erection, although it may be doubted whether the ordinary cast iron sections are as easily cleaned as is the pipe surface, properly arranged, and they certainly take up more space.

The velocity of air generally allowed through a battery is from 12 to 18 feet per second with a centrifugal fan and from 8 to 12 with a propeller type.

The condition of accessibility implies the provision of sundry baffle plates and access doors, so that the air cannot pass from the fan to the flues without being forced over the battery surface. Otherwise the efficiency of the battery may be decreased.

The Mixing Chambers.

The mixing or diffusing chambers in which the air is tempered ready for delivery into the flues must be so arranged that as far as possible the air passing through the battery has a uniform velocity all over, and that the air passes into the flues at a uniform temperature, so that streams of hot and cold air do not exist side by side in the flue, that is, the warm air and cool air are thoroughly mixed.

In order that it shall be possible to regulate the temperature in the flue within wide limits, and in as short a time as possible, it is of course necessary that it shall be possible, by means of baffle plates, to short circuit some or all of the air from the fan, so that it may be directed into the flues without going through the battery at all.

Regulation of temperature can, of course, be to some extent effected by means of valves on the heating surface. This method of regulation involves trouble, and expense and time, for the battery does not cool down as rapidly as is desirable when it is required to make some alteration in the temperature of the air being delivered into the flues. In any case the alteration in temperature of the air in the flues is only slowly effected, especially where they are constructed of brickwork, on account of the heat in the substance of the walls.

It is desirable that a mixing chamber shall be provided immediately before the entry into the flues. This chamber can also be used for a humidifying chamber. The cold air should enter this chamber at the top. If it enters at the bottom there is a danger of stratification of the air in the flues into layers at different temperatures. If this effect is produced those outlets which are near the fan will deliver sometimes air which is too hot and sometimes too cold.

All flues, especially those which are at all exposed, should be very carefully protected against loss of heat. A very large annual loss may be caused if this provision is neglected, besides which, the effect of neglect is that the air delivered at the end of the flue is very distinctly cooler than that delivered near the fan, and this tends to inequality of result.

The trunks are generally formed either of brickwork or of galvanised sheet metal, according to their position. The latter when properly constructed cause much less friction than do the former. In all cases, sheet metal trunks, if made with lapped joints, should have the edges of the plate inside the flue pointing forwards in the same direction as the air travels, otherwise quite a large increase of friction is caused.

In the case of brickwork, all joints should be made as tight and as smooth as possible, and it is very desirable that the inner face of the brickwork should be glazed. Ordinary rough brickwork, however carefully cleaned, is apt to accumulate dust and dirt to a much greater extent than is a glazed surface.

Each branch outlet should have a permanent regulating damper which can be adjusted by means of a set screw, so that it can readily be altered.

These adjustments are necessary because it is impossible so to calculate the outlets as to give a correct distribution of air, without at the same time making many of them so small that they cannot be cleaned.

It is in all cases desirable that the branches leading to the individual rooms shall have so high a resistance, natural or artificially produced by regulating dampers, that a distinct and appreciable pressure is maintained in the main itself. If this is not done, the distribution of air cannot possibly be uniform.

In certain states of temperature the absence of pressure in the main trunk will cause air to be syphoned out of one room into another, especially as a flow of air of any considerable velocity up one flue tends to have an induction effect, so that it is no uncommon thing for the pressure towards the end of a badly designed flue to be actually slightly below atmospheric, that is, that there is a suction in the flue. When this is the case the most remote flues will not be inlets at all, but will abstract air from the corresponding rooms.

In order to avoid this, it is necessary not only that the fan shall be well above its power, both as to pressure and volume, but also that the main flue shall be so large that the positive pressure is maintained in it throughout its whole length. Only in these circumstances can uniformity of result be obtained.

Similar remarks in regard to the relation between the sizes of the individual extraction outlets and the main extract trunk apply, as in the case of the pressure trunk, that is to say, where a fan is employed for the extraction the individual flue must have so high a resistance that a vacuum is maintained throughout the whole length of the main trunk. That vacuum will, of course, be greatest near the fan, and the resistances of the individual flues must be regulated accordingly, either by making them of suitable size or by controlling their resistance by means of fixed adjusting dampers.

Extraction Outlets and Flues.

In some of the cheaper installations these are entirely omitted, and the air is left to find its way out as best it can, through cracks in windows, walls, doors, etc. Where these are of fairly tight construction, this often leads to the depressing of the neutral zone below the floor level, and in so far is an advantage in suppressing primary draughts. But this omission cannot be defended on any other ground.

The amount of air which will be passed through a room depends then entirely on the *amount* of leakage, and is essentially uncertain, and cannot be determined to within 50%.

The opening of a door may cause a temporary strong current of air which will be felt as a draught either inside or outside the room, or both.

A further grave disadvantage has been already alluded to, and is, that the internal pressure set up in the room causes the walls to be permeated with vitiated products and dust, etc., which causes smells, which may be very unhealthy.

Outlets into the open air from a room are almost equally undesirable because changes in the wind outside may cause such outlets to become inlets of cold air, which will certainly give rise to complaint of draught, however powerful within reason the fan may be. The position of such outlets has already been discussed in the chapter on the movements of air in a room.

For satisfactory results, therefore, it seems necessary to provide a system of outlet trunks. If each separate trunk is connected to the open air, there is still liability to their becoming sources of cold inlet draughts, especially when the fan is not at work.

The alternative to this is a collection of these flues into the common trunk in the roof. This again gives rise to the possibility or probability of undesirable accumulation of dirt in the flues, which are very difficult to clean.

Where such flues are provided there is the further possibility of the transmission of sound from one room to another through them, or of air syphoning over from one room to the other, unless the flues are provided with an extraction fan, which will maintain a certain degree of vacuum throughout the flues. This again adds to the expense.

If the common trunk is made of wood, since this will generally be distinctly colder than the room when it is occupied, it is very apt to rot, owing to the dampness which will in consequence be deposited from the extract air in cold weather when the rooms are crowded. This leads to rotting of wood, so that metal flues are desirable in this position.

Wooden flues are also especially dangerous in case of fire, as they provide an easy path for the fire to travel.

The Calculation of Quantities in the Plenum System.

The principles governing the calculation of quantities of air and moisture and the heating and humidifying of the air have been discussed in Chapters IX. and XXI.; the necessary power of the fan in Chapter XI.; the principles governing the loss of head in flues, owing to the passage of given quantities of air, in Chapter V.

It remains only to explain the application of these various principles to the special case where the Plenum System is adopted. The sizes of flues can only be calculated by a similar process of trial and error to that which has already been explained in connection with the sizes of pipes for low pressure hot water and low pressure steam installations.

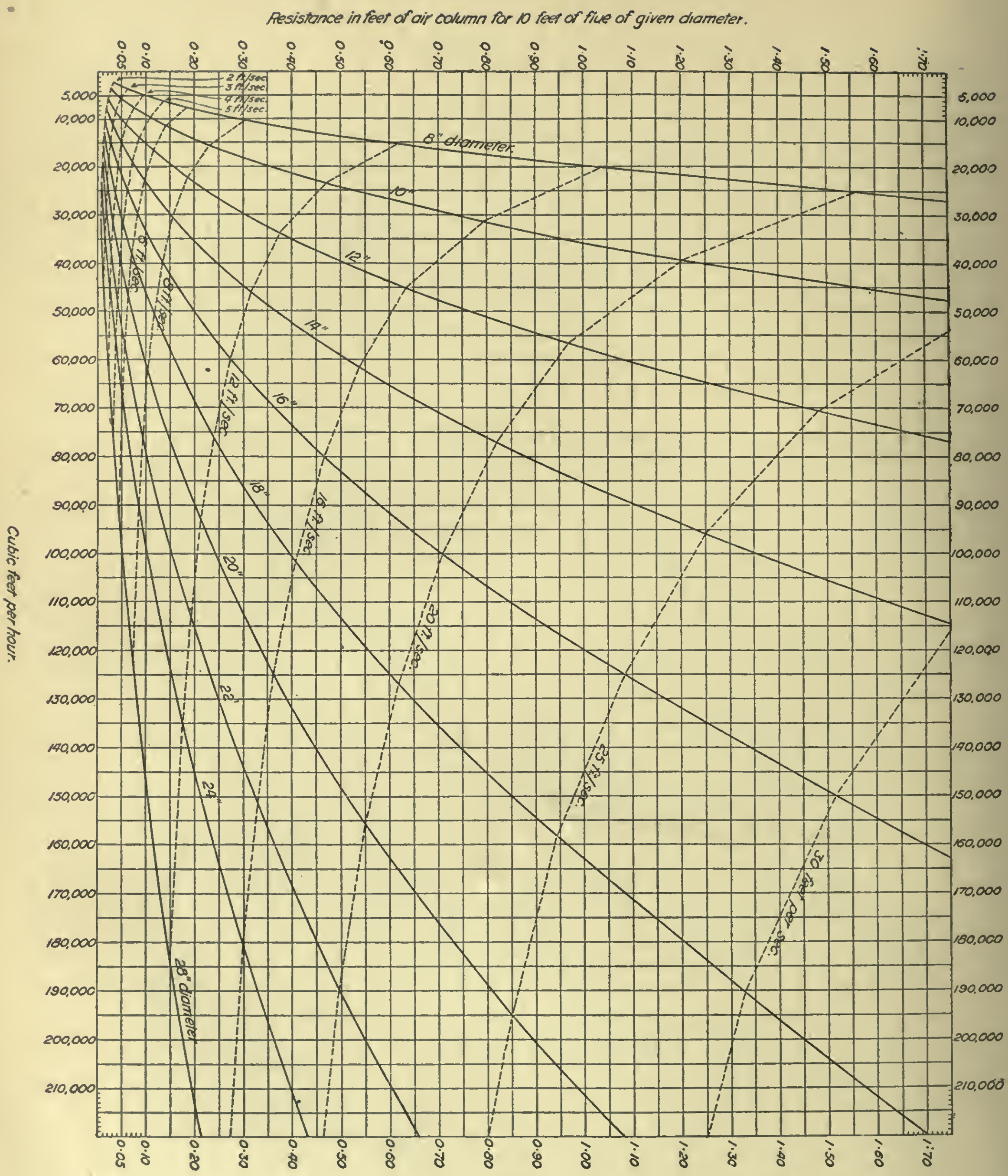
A complete drawing of the entire system of flues, preferably in isometric projection, must first be made and the quantities of air required and their temperatures marked on each portion of the flue.

The pressure head produced by the working of the fan should be accurately known, and this pressure must be first converted into feet of air column at the same temperature as that of the air in the flues. Suitable choice must next be made of the pressure which it is desired to maintain at the last junction most remote from the fan. This should be generally not less than ⅛″ of water, and in many installations preferably ¼″.

The size of the last two circuit flues (so called by analogy with the circuit pipe leading to a hot water radiator, as to which see page 467) may next be determined. This can be readily done by means of the diagrams given, which shows loss of head per 10 foot run in flues of various sizes.

The flue should be calculated of such a size that when the required quantity of air is passing through it the pressure at the grating inlet to the room should be atmospheric. Suitable allowance must of course be made for access of pressure due to changes of velocity.

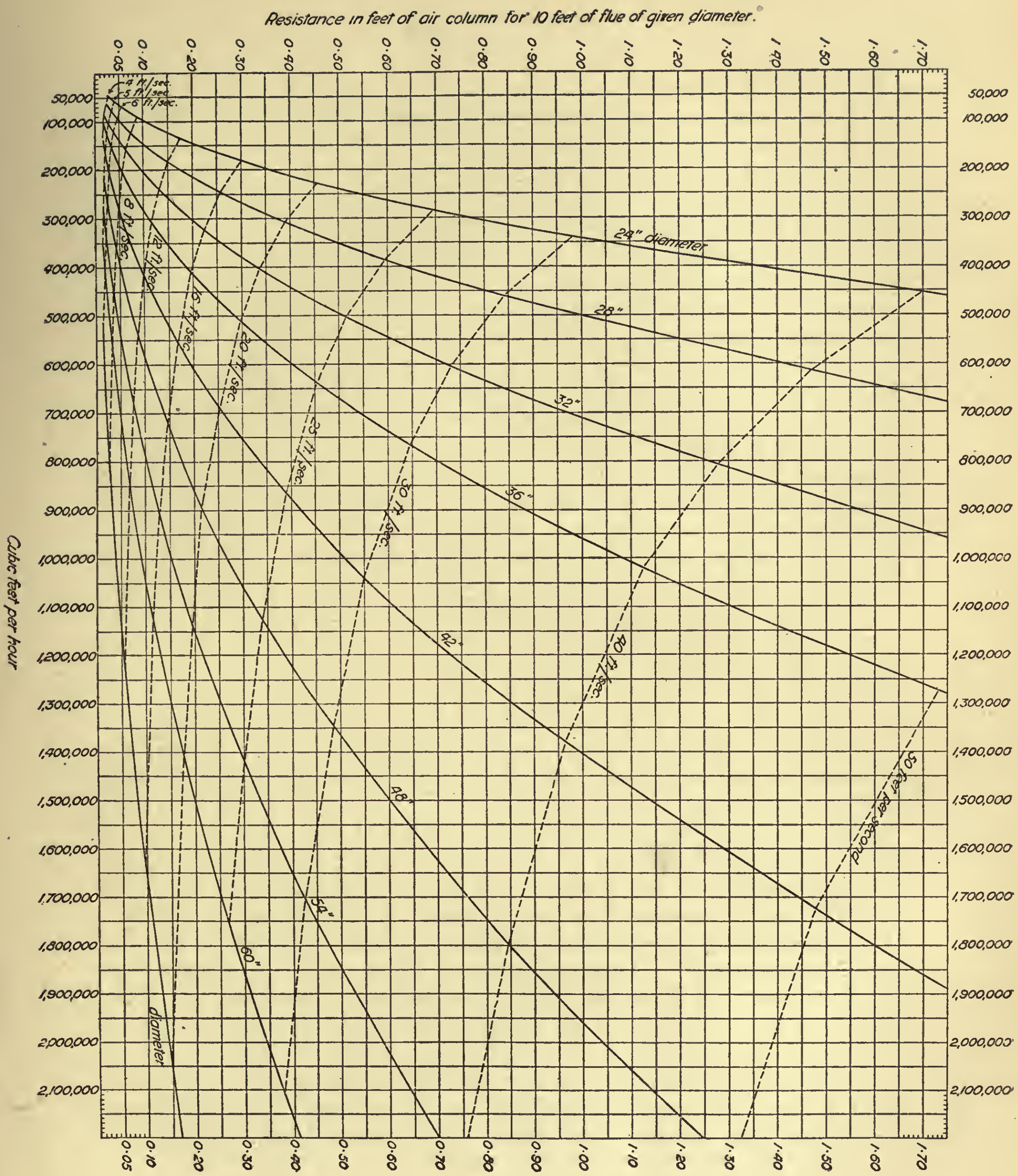
At the same time the size of the grating should be such that the velocity of delivery into the room should not be greater than 5 feet per second, except in special cases. A higher velocity than this will cause trouble in regard to draughts. This requirement implies that the number



FLOW OF AIR IN ROUND IRON FLUES WITH FRICTION IN FEET OF AIR COLUMN CALCULATED FROM WEISBACH'S FORMULA WITH RIETSCHEL'S COEFFICIENTS.

A. H. Barker, delt

FIG. 139.



FLOW OF AIR IN ROUND IRON FLUES WITH FRICTION IN FEET OF AIR COLUMN CALCULATED FROM WEISBACH'S FORMULA WITH RIETSCHEL'S COEFFICIENTS.

A. H. Barker, delt.

FIG. 140.

of square inches of grating per thousand cubic feet of air delivered per hour should be at least 8, or one square foot for 18,000 cubic feet per hour, from which the size of gratings can be immediately calculated.

It is commonly necessary that each circuit flue shall be provided with an adjustable damper which will introduce the required resistance into the flue, otherwise the flue would in usual cases be too small to be cleaned. It is often advisable to reduce to a considerable extent the diameter of the flue for a short distance near the outlet from the main trunk, so that the pressure in the main trunk may be maintained and a uniform distribution secured.

The approximate sizes of all main trunks may be determined according to the degree of silence required in the installation and the power available. A very high velocity in the flue is apt to cause a noise, especially in buildings in which a normal degree of quiet is necessary. Indeed, with most Plenum systems it is possible to be conscious of a certain noise of movement which may be more or less subdued according to the perfection of the design.

It is the experience of the present Author that this noise will not give rise to the complaint if the velocity of the air in the main flue does not exceed 20 feet per second. The greater the velocity above this point the more is the noise appreciable. Lower velocities than this are commonly desirable where they can be secured, both on the ground of economy of upkeep and in the interests of silence. But the conditions vary so much that no hard and fast rule can be laid down.

Whatever velocity is provisionally determined on in the flue, the corresponding pressure should be calculated back towards the fan, at each junction or each reduction of area.

All the circuit flues leaving the main trunk at this point can next be calculated so that when the given quantities of air are passing through them the pressure at the grating shall be zero, allowance having been made for the alteration in pressure due to variations in area in the flue. This calculation will determine the pressure at the entrance to the flue nearest to the battery. To this pressure is to be added the resistance of the various parts of the central apparatus, including the battery and the screens. The arrangement of these varies so greatly that no general information can be given.

Commonly a quarter of an inch of water pressure or its equivalent in air column is found to be absorbed by the passage of the air through the battery and screens. The figures which have been given in this connection should suffice to give the engineer an approximate idea of the loss of pressure which may be expected, but no exact accuracy can be looked for in these figures otherwise than by actual experiment on the type of apparatus employed.

General Remarks on Design of Flues.

It will be seen by an inspection of the table of resistances to what a very great extent the efficiency of a system of flues depends on the freedom of the design from sharp bends and sudden changes of direction.

It will be seen by careful comparison of the figures that the resistance of several hundred feet of straight flue is actually less than that of one single sharp elbow or bend, and that in consequence it is impossible to take too great care that sharp elbows and sudden changes of direction are absolutely eliminated from the design. This requirement is commonly extremely inconvenient and expensive, and there is large room for ingenuity in avoiding any such sudden changes.

A change in direction having a radius of five times the diameter of the flue is commonly sufficient to avoid any considerable loss, but the accommodation of a flue of large radius in the structure of a building is frequently a matter of very great difficulty, and a really good design can commonly only be carried out by close co-operation between the architect and the engineer before the actual structure is commenced.

CHAPTER XXXII.

ON HEATING BY FIREPLACES AND STOVES.

ONE of the simplest of all methods of heating consists of what has been called in an earlier portion of this work the method of "direct heating." This consists in burning the fuel from which the heat is derived in the apartment to be heated. This method is, strictly speaking, only direct—that is to say, the heat is only obtained directly from the burning fuel—in the case where the fire is open to the air of the room.

In a certain sense when the fire is enclosed the heating becomes partially indirect, because the heat is then derived not directly from the fire itself, but from air warmed by conduction through the walls of the stove.

The direct method of heating is probably used in England more widely by far than all other methods combined. It appears doubtful whether the open fire, at any rate, will ever be banished from popular favour here, because of certain inherent qualities not possessed by other methods which this method of heating has. In particular allusion is made to the cheerfulness of an open fire. It may be doubted whether any person who has been accustomed to the glow of an open fire ever becomes fully reconciled to any other method of heating, at any rate for living rooms.

The open fire has obtained a greater popularity in this country than in continental and American countries on account of the milder climate to be found here.

One leading advantage of the open fire which has been previously alluded to, is that it provides a powerful means for ventilating the room in which the fire is burning. The ventilation of a room in which a fire is lighted is commonly 300% or 400% better than when the fire is not lighted. Fireplaces are commonly made with an open throat so that the temperature of the hot gases proceeding immediately from the fire is greatly reduced by the large quantity of cold air drawn into the flue from the room without passing through the fire. The temperature commonly found in such a chimney flue after the admixture of the cold air is from 100° to 150° F., sometimes in extreme cases reaching 200°, but rarely more.

An open fire has certain subsidiary advantages not connected with its heating effect, but with the general convenience and habits of the persons occupying the room. Such apparently trifling features as forming a convenient receptacle for cigar ends, tobacco ashes, matches, and waste paper, although apparently absurdly inadequate as a reason for the popularity of the open fire, doubtless have more than their proper share of weight in establishing the favour with which this method of heating for living rooms is almost universally regarded in this country.

The great majority of rooms in this country are not warmed during the winter months. It is only a small proportion of bedrooms in private houses, for instance, which the occupants find it necessary to warm in winter, and only in cases where the occupants are either of luxurious habits, invalids, or very aged persons. It is the prevailing impression here that a warm bedroom is not conducive to health. To provide radiators in bedrooms for such occasional services would involve more expense than is thought to be justified in this country. For these purposes an open fire can often be purchased at a small cost and built into the wall provided with a chimney. This is used if and when occasion requires.

In very cold countries, on the other hand, some form of heating in the sleeping apartments is necessary throughout the great part of the year. In such climates there is a much greater inducement to adopt central heating apparatus than here.

The disadvantages of an open fire as a means of heating are sufficiently obvious. It is extremely extravagant in fuel. The efficiency is commonly not more than one-fifth or one-sixth of that of an ordinary system of heating by radiators for an approximately equal effect. It causes an immense amount of labour in carrying fuel and the removal of ashes. In addition it produces a great deal of dust, a grave disadvantage from the domestic point of view.

With all these very salient and admitted disadvantages the open fire retains its favour and is probably as much in favour to-day as it was before the method of central heating was invented.

The character of the heat emitted by an open fire is entirely different from that given off by a radiator, so different, indeed, that it is difficult and almost impossible to institute a fair comparison between them. The greater portion of the heat emitted by an open fire comes into a room in the form of radiation direct from the glowing coals. Another portion reaches the air of the room through the medium of the brickwork surrounding the fire, which itself becomes warmed in consequence of the high temperature of the fireplace and so warms the air of the room partly by convection currents and partly by producing a feeling of warmth from the largely exposed surfaces of warm wall.

The character of the heat emitted as radiation is entirely different from anything produced by a radiator or any other form of heat, namely, high frequency etheric rays. The heat given off as radiation from a radiator consists of rays of relatively long wave lengths. That from an open fire consists of heat and light rays of relatively short wave length. The greater portion of the heat is given off from a radiator as convection currents of heated air. Speaking strictly, a radiator is misnamed. It would be more logically called "a heat convector."

A room heated by an open fire is filled in every part with these high frequency heat waves, some of which impinge on people in the room. If not intercepted by the body of one of the occupants or by some piece of furniture all rays reach the walls of the room, where they are partly reflected and partly absorbed.

It will be seen that no adequate comparison can be made between the effect of the energy so given off and that emitted by a radiator. The latter is comparatively easy to measure in B.T.U. Whether or not it is given off as radiation or convection, suitable observations of the quantity of water and the difference in temperature between the flow and return will give a closely accurate estimate in British Thermal Units of the total energy actually supplied to a room by a radiator.

An accurate measurement of the amount of heat given to a room by an open fire on the other hand would be so difficult as to be practically impossible. It could only be arrived at by a series of elaborate experimental observations involving the use of a great deal more plant and other preparation than has yet been devoted to this investigation.

Even assuming that it is known of a certain open fire maintained in a certain condition, that it communicates to the room on the whole the same amount of energy as a radiator also fixed in the room, it is clearly impossible to equate the two effects. It must be ascertained how much of the energy passing through the room as radiation finds its way to the outside of the building either by direct radiation through the glass or by absorption at the surface of the wall and subsequent conduction through its substance, and so on to the outer air.

It is known that glass is highly diathermanous to heat rays proceeding from a high temperature source such as an open fire, but exceedingly opaque to rays proceeding from a low temperature source, such as a radiator. Any portion of the rays from the open fire, therefore, which fall on the window glass are transmitted as radiation to the outer air and so are absolutely lost.

It is notorious that rooms with large windows cannot be adequately warmed by open fires immediately opposite them. The rays proceeding from a radiator, however, are absorbed by the surface on which they impinge. Some of them are reflected back into the room from that surface. The absorbed portion of the energy of the radiation passes through the glass by conduction only.

As previously stated, a large portion of the heat emitted by the radiator is given off as convection currents of warmed air. These circulate to all parts of the room and so maintain a moderately uniform air temperature throughout.

A shielded thermometer hung in a room heated by the radiator would indicate a much higher temperature than if the same amount of energy were emitted by an open fire. It has been pointed out in Chapter XVII. that a thermometer hung in a room does not indicate accurately the temperature of the air, since it is affected by radiant heat which is independent of the air temperature.

A thermometer having a blackened bulb always gives a different reading from one having a polished bulb when hung together in the same room through which radiant heat from a source at a higher temperature than the air is passing, because the former absorbs all the radiation which falls upon it while the latter reflects the greater part of such radiation.

It will thus be seen that any logical or scientific comparison between the heating effect of a radiator and an open fire is extremely difficult, if not impossible. The ultimate criterion of the value of the heating from the point of view of comfort is a subjective effect which is almost incapable of scientific investigation.

The sight of a glowing fire in a cold room gives a feeling of comfort and warmth which has no counterpart in the case of a heated radiator. The fundamental object of warming the room is to produce this feeling of comfort and not to produce an effect on the thermometer reading. The use of the latter instrument is only a somewhat crude and uninspired method of endeavouring to reduce subjective sensations to a numerical scale.

It is probable that a suitable combination of a shielded ordinary thermometer with a black bulb thermometer might form such a means if combined with a very careful series of observations on a number of sensitive subjects, capable of an introspective analysis of their own sensations. Such an instrument if properly constructed and used would give at the same time the true temperature of the air combined with an accurate measure of the effect of the existing radiant energy. The matter is in urgent need of investigation, having regard to the great importance of the open fire as a means of heating in this country.

There is no doubt that the true temperature of the air in a room is responsible for a certain portion of the effect in producing a feeling of warmth, but exactly what proportion is not known. At any rate, when the air temperature in a room is below a certain minimum, which varies for each individual, a feeling of cold is unquestionably produced in a person at rest within the room, in spite of very powerful radiation which may be passing through the room at the same time.

No reliable method has yet been devised for estimating the degree of the feeling of warmth in a fire heated room.

If, on the other hand, however, the air is too warm, a feeling of oppression is produced. The effect of the radiation is to warm those parts of the body of a person exposed to the rays, and the chilling effect of the cold air on the other portions not directly exposed to the radiation is the cause of the complaint made against the open fire in cold weather, that people are boiled on one side and frozen on the other, which complaint has certainly considerable justification.

It appears, therefore, that if no regard is had to expense in first cost or of upkeep, a combination of open fire with radiators is most to be desired for sitting or living rooms. The latter have for their main object the maintenance of the temperature of the air just above such a point as will cause a feeling of chill in those parts of the body of persons not exposed to the direct action of the rays.

For these reasons it is almost impossible to estimate the effect of an open fire by figures which have any meaning or which have any bearing upon the problem of the degree of comfort produced in the room heated. That effect depends not only on the fire itself, but also on the chimney piece and the outside surface of the flues above and around the fire. If these are of such a design as to maintain the wall above and around the fire at a good temperature the effect of the fire may be greatly increased.

The number of thermal units emitted from a brightly glowing fire in the form of radiation has been estimated by the Author to be about 3,500 to 4,000 B.T.U. per square foot of projected fuel surface. This figure is, of course, very uncertain because the amount of radiation varies enormously according to the temperature of the combustion.

In order to maintain this degree of heat it is commonly necessary to burn from 3 to 3½ lbs. of fuel per square foot of projected fire surface per hour. If this heat is the only heat available, in estimating the effect of an open fire and on the assumption that the coal has a calorific value of 13,000 B.T.U. per pound, this would give an efficiency of about 10% as the figure of merit of the open fire as a means of heating by direct radiation alone.

This radiant heat is, of course, given off in all directions equally. A fire as ordinarily constructed is placed in a recess or chimney piece in a wall in such a way that in no case can more than one half of the total radiation come into the room directly from the fire. A large proportion of the radiation passes direct up the chimney where it is absorbed by the smoke and soot and so the heat energy is carried to the outer air along with the flue gases and lost.

One of the very few fundamental improvements that have been made in open fires consists in the trapping of some of this heat and projecting it into the room by the provision of fireclay slabs at the back of the fire leaning forward in such a way that much of the radiant heat passing upwards is caught by the sloping slabs and either reflected or absorbed and re-radiated into the room from them.

This device normally increases the radiant heat from a fire by 20 or 30%.

Other improvements consist in the provision of an opening admitting either fresh air or room air to the space at the back of the fire, whereby some of the heat transmitted by conduction through the plaster or slabs at the back of the fire is communicated to the entering air, which finds its way into the room through gratings provided above the fireplace. By this means the air is warmed by conduction as well as the surroundings by the radiant heat.

There are few or no reliable experimental data on the effect of open fires, owing perhaps partly to the want of understanding on the subject and partly to the extraordinary difficulties of experimenting.

Some few years ago a set of experiments were made under the auspices of the Coal Smoke Abatement Society with the object of testing a large number of makes of open fires submitted for approval in a large Government contract for the warming of a block of office buildings by fireplaces. Particulars of these trials are given in the "Lancet" of May 19th, 1906.

Their results are of little value on account of the way in which they were made. The method consisted in fixing each of the separate grates in different rooms and burning measured quantities of coal on the grates. The only observations made with a view to obtaining a balance sheet of the energy contained in the fuel was to measure the temperature and volume of flue gases passed per hour up the chimney. The "efficiency" was calculated as the proportion of the heat given to the air passed through the room per hour to the total heat in the fuel. The unobserved and unmeasured balance, which consisted of from 80% to 90% of the total heat, was passed over as being "the percentage of the total heat of coal burnt which is given to the walls, etc., of the room chiefly by radiation."

An experiment in which the remainder, consisting of wholly unobserved factors, amounts to 80—90% of the total heat used, cannot be seriously considered as of any value.

It need only be pointed out that according to the results of these experiments some 39,000 B.T.U. were communicated to a room whose contents were 3,900, and the effect was said to be that the temperature of the room was raised about 16°. Such a result conveys its own refutation.

No particulars are given of the dimensions of the heat losing surfaces. During these tests the average value of the temperature of the flue gases varied from 84° to 100°, the fuel consumed varied from 2½ to 3¾ pounds per hour, and the flue gases varied from 6,300 to 11,000 cubic feet per hour, or from 2,000 to 3,500 cubic feet of air per pound of coal. This is about ten times the quantity of air that would normally be employed in a boiler furnace.

The heat lost in the flue gases in these experiments was from 4,000 to 7,700 B.T.U. per hour, or about 13½% of the total heat, though as no particulars are given of the method in which these figures were obtained it is impossible to criticise them.

The chimney flues were 11″ diameter by 68′ high. The velocity of the smoke gases up the chimney therefore appears to have been from 2.65 to 4.65 feet per second, which is a much lower velocity than might be expected, and appears to suggest some error in observation.

The calculated smoke velocity in a flue of this height maintained at this temperature would be about 8.5 to 9 feet per second.

These experiments are only quoted as an illustration of the futility of endeavouring to derive any useful information by what may be called the "practical" method.

Stoves.

A very large increase in the apparent efficiency of a fire is brought about by enclosing it within a stove, which is placed in the room to be heated a short distance away from the wall up which the flue passes. Such stoves have attained a certain amount of popularity in this country for heating, where low first cost is an essential. They are much more widely used in America and on the Continent, where some form of heating appliance is much more of a necessity than in this country.

They are made in a great variety of different patterns. The surface of the stove is kept hot by the combustion proceeding within and communicating heat by conduction through the outer casing to the air of the apartment, partly by radiation.

Their use is not generally popular in this country on account of the oppressive character of the heat which they produce. They are, however, much more economical than the open fire and they may even show a fair efficiency if provided with a small fire and a large amount of heating surface. Up to 40 or 45% efficiency is not uncommon, though the majority of stoves do not exceed 25 or 30%. It is the custom for the manufacturers of these appliances to specify the cubic contents of the room which a given stove will heat, this being, of course, a relic of the old rule of thumb which proportions the amount of heating surface required by a room to its cubic capacity. It is altogether unsatisfactory as a reliable basis.

The amount of heat required by a room depends, of course, on its exposure factor and its interchange of air and not on its capacity, except in so far as the latter affects the amount of air passing through the room. The amount of heat given off from a stove depends, of course, on its surface temperature, and this cannot generally be exactly known, depending as it does on the intensity of the combustion within the stove and on the conductivity of the fireclay blocks with which it is lined.

The surface temperature of a stove is commonly 300°—400° F., though 800° to 1,000° is not unknown with common stoves, indeed, the surface of some stoves is commonly found to be at a dull red heat.

The usual figure which may be employed for calculations for whatever purpose is that each square foot of stoves emits about 800 B.T.U. per hour. As before stated, the efficiency being about from 30% to 40%. According to these figures, if a stove is required to emit, say, 6,000 B.T.U. per hour, it should have in ordinary circumstances a surface of about 7 or 8 square feet and with fuel having a calorific capacity of 14,000 B.T.U. per hour it should consume less than 1 lb. of fuel per hour. This would generally suffice for the adequate warming of a room of 3,000 cubic feet.

Stoves for the combustion of wood and peat as fuel are largely manufactured on the Continent, but do not find much employment here on account of the abundance of coal fuel on the one hand, and the scarcity of wood and peat and similar fuels on the other, and also for the above named reason that a large majority of the rooms in this country are not heated at all in the winter on account of the relatively mild climate.

CHAPTER XXXIII.

ON TESTING OF PLANT FOR HEATING AND VENTILATION.

General Considerations on Practical Tests.

To obtain a satisfactory practical test of the kind of plant with which the heating engineer has chiefly to do is perhaps as difficult a problem as commonly occurs in the whole of engineering practice. The causes of the difficulty have been generally indicated throughout the previous chapters of this work.

The transference of heat from one body to another is one of the most subtle phenomena known to science. There is nothing to see when heat passes. The full effect of its passage from one body to another is very difficult to determine. Measurements of its quantity are in their very nature indirect only, and are frequently not only indirect in the first degree, but in the second or third degree. The quantity of heat measured is deduced from a phenomenon which is itself caused by some effect of the passage of the heat. In consequence accurate measurements of its quantity are in many cases almost impossible.

As a contrast consider, for instance, measurements of quantities of electrical energy. Such measurements can be made almost directly by means of suitable instruments. Although these operate in a manner which is really indirect, yet the observations are only indirect in the first degree. In effect they do accurately measure the quantity of electricity as it passes.

The measurements of heat movements involve the time element to a large degree as a separate observation, which is not the case with many other phenomena.

One of the chief difficulties arises in almost all cases from the circumstance that it is impossible to co-ordinate the conditions under which the observations are made with those which will prevail in practical work.

A test is of no considerable scientific value unless it is made with such precisely defined conditions that it can, when necessary, be repeated with substantially the same results.

Suppose, for instance, the amount of heat transmitted through a plate is in question. A test on one occasion gives a rate of transmission 300 B.T.U. per square foot. On another occasion, when the same test is made, the result is, say, 445 B.T.U. per square foot, and on another occasion 225, and so on, it is obvious that from widely different results such as this no definite information can be obtained, except something like the following:—

In unfavourable conditions the heat transmission is about 200 B.T.U. per square foot, rising when the conditions are favourable to about 500 B.T.U. per square foot.

The criterion of the value of a test from the point of view of the scientific engineer is that it shall be made in such a way that the results can be exactly repeated or reproduced when occasion requires, and that it shall therefore enable results of future experiments to be foretold with some accuracy.

If the magnitude of a result depends on the value of several conditions, unless each such condition is so regulated that it has the same value during each test, the result cannot be the same.

A *complete* test of any phenomenon implies that not only shall it be possible to repeat any particular result by controlling the combination of conditions to the same degree on each occasion, but it must also reveal what is the effect on the result of altering each of these conditions between the maximum possible and the minimum possible for each separate condition on which the result depends.

Even this great complication is not the only difficulty. It sometimes, indeed frequently, happens that the complete result of a test depends not entirely on the separate individual values of the conditions, but on their values in combination.

Suppose, for instance, the magnitude of a certain result Z depends on the values of four conditions, A B C and D, and that one result Z_1 can be obtained as often as desired by maintaining each of the functions A B C and D at particular values A_1 B_1 C_1 and D_1. The result Z varies from the value Z_1 every time any of the conditions A B C or D varies from the values A_1 B_1 C_1 D_1.

Suppose that B C and D being kept constant at values B_1 C_1 D_1, A is allowed to vary between its minimum and its maximum, observations of the value of Z being taken for a number of separate values of A. A curve can then be drawn showing the values of Z corresponding to each value of A, *when B C and D are respectively kept constant and at one particular value for each variable.*

Similarly, when A C and D are kept constant and B is allowed to vary, another curve will be obtained, and so on for the other variables.

But it must be noticed that each of these curves obtained for the variation in the values of Z assumes that three of the variables A B C and D have the particular values which were assigned to them during the experiment.

The question then arises, what will be the value of Z when, for instance, A and B vary in combination, or when A B and C vary in combination, or when A and C vary in combination?

It will be seen that when, as is so frequently the case, the value of Z depends on the values of several independent variables, any one of which may vary separately or in combination with other variables, the result becomes almost of immeasurable complexity. In this case any result that can be obtained cannot be regarded as accurate, unless the entire field of possible combinations has been fully surveyed, otherwise the results cannot be taken as applying with accuracy to practical conditions. In practical work such a result is only of value when it enables the engineer to predict with some confidence what a particular result will be when the values of the different variables are exactly specified.

If the result depends on factors the value of which the engineer does not know, it is evident that he cannot predict the result with certainty.

Now the value of many or most of the results which the heating engineer endeavours to obtain depends on at least four or five independent conditions.

For instance, the heat lost Z from a room with walls of given thickness and known material, and having windows, ceiling, etc., of known size and normal construction, is known to depend on (A) the difference in temperature between the inside and outside air, (B) on the amount of ventilation passing through the room, (C) on the velocity of the wind, (D) on its direction, (E) on the humidity of the air, (F) on the dampness of the walls and on other conditions.

For the purposes of this discussion we shall leave out of account the almost immeasurable complexity introduced by the regenerative action of the walls, explained in Chapter XIV.

With a given wall condition and given weather conditions and given rate of ventilation, the heat lost depends on the difference in temperature between inside and outside. It would be possible to obtain a curve showing what the heat lost Z would be when the temperature difference between inside and outside varies. This may be indicated as the curve A.

It would also be possible to conceive that a curve might be obtained showing the amount of heat lost with a given difference between inside and outside, when the air interchange alone varies. For instance, with a constant outside temperature of 30° and an inside temperature of 60°, the loss of heat when different quantities of air per hour are passing through the room may be determined.

Exact tests do not support the common supposition that loss of heat by air interchange can be calculated independently of the loss of heat through the walls. The additional heat which must be supplied to a room to make up for an additional thousand cubic feet of air per hour passing through it, is not apparently so great as the amount of heat necessary to raise one thousand cubic feet of air from the outside to the inside temperature. The reason for this apparent anomaly is that the reservoir of heat retained in the walls, derived from the combustion of fuel which has taken place previously to the observations, is drawn upon by an increased quantity of air and vitiates the observations.

A curve may therefore be drawn showing the total value of Z the loss of heat, when different quantities of air are passing with a constant temperature difference of 30°.

The question now arises if the difference in temperature between inside and outside becomes 35°, what will be the nature of the curve showing the values of Z for different quantities of air passing through the room?

Consider now a further factor. Assume a given wall condition, and a given rate of ventilation, and a given temperature difference between inside and outside of 30°. Let the velocity of the wind alone be varied. A curve might thus be obtained showing the loss of heat Z for all velocities of the wind from zero up to, say, 20 miles an hour, with a given temperature difference of 30° and a given air interchange.

This curve having been obtained, the question arises, if the difference in temperature between inside and outside becomes 35° and if the amount of ventilation changes from the given value (as it certainly would change in practice), what is the effect on the heat lost?

It will easily be seen that this double alteration of conditions throws the whole problem again into the region of uncertainty. To solve it with scientific accuracy it would again be necessary to make a series of tests, if such were possible, maintaining the difference of temperature at 35° and with the increased interchange, and varying the velocity of the wind. No formula having even a remote resemblance to scientific accuracy could be evolved from such a miscellaneous group of experimental results, as could conceivably be obtained by observations on practical conditions.

It will easily be seen that every additional cause of variation of heat loss increases the complexity of the problem greatly.

The time element also has to be taken into consideration, the outlines of which problem have been indicated under the head of "time-lag" in Chapter XIV.

In just the same way the heat lost from a radiator varies, according to at least as many of these conditions. The temperature effect within a room produced by radiators placed within it, therefore, depends on a multitude of considerations. For each particular condition of the radiators has to be combined with possible different conditions in the room. It is not difficult, therefore, to see that any solution of the problems of the heating engineer, which is both simple and exact, is an entire impossibility. All solutions are in their very nature approximate only, and can never be anything else.

"Practical" Tests.

A good deal of misconception exists in the minds of practical men as to what constitutes a practical test, and this misconception leads to a good deal of foolish talk about the relations of the scientist and the practical man, "theory and practice," and so forth.

A test is not a practical one merely because it is an ignorant one, neither is it made more practical by neglecting observations on points which are known to exercise an important influence on the result.

From the point of view of the practitioner the experimental results to be desired are the simplest which will enable the practical effect of any given combination of parts to be predicted with certainty. If that effect depends on a number of different factors it is quite certain that a correct final result cannot be determined by calculations unless the corresponding value of each of these factors has been taken into account.

It is mere foolishness to shut one's eyes to this self-evident fact, and to condemn the attempt to take account of such factors as "theory," as is so often done by the "practical man," who often fails to understand the real fundamental nature of the problem with which he has to deal.

From a practical point of view the nett result of all these variations is summed up in their joint effect on the thermometer. It will therefore be seen that a limit is soon reached beyond which, from the point of view of the practical man, it is a mere waste of time to carry calculations or to make tests with the object of securing accuracy. It is impossible for the practical man to afford the necessary time. It pays him better to allow a margin to cover all such deviations.

The point of view of the practical scientific investigator is quite different from that of the "practical man." The former endeavours to obtain results which can be repeated by controlling exactly the attendant conditions on which the value of the result depends and to express the result in terms of the observed values of the conditions.

The busy contracting engineer frequently requires results given to him in one figure, which in their very nature cannot be expressed by one figure.

Consider, for instance, a test of a steam tube heater. A practical man would desire to know if he puts a certain length of tube into a volume of water how much heat will be conveyed per hour to the water. The actual quantity conveyed depends entirely on the velocity of the water surrounding the tubes, and on its temperature, on the perfection with which the steam coil is drained, and on its state of cleanliness.

To give one figure in such a connection would be wholly misleading. To give more than one figure is frequently very embarrassing for the practical man who cannot specify his conditions and does not know what the velocity of the water surrounding the tubes will be, nor what will be the mean temperature nor quality of the water, nor the value of the other factors on which the result actually depends.

Similar questions arise in practically every case.

In order to control conditions and obtain precise information regarding the result of their variation, the scientific experimenter has frequently to make tests under conditions which are at variance with those under which the apparatus will be used in practice.

The test may, of course, be of little practical value if these conditions are widely different from those which prevail in practical work. The latter are essentially indeterminate and variable, the former must as essentially be maintained precise and constant.

It follows that tests carried out in such conditions as will satisfy scientific requirements when viewed in the light of practical application sometimes appear almost absurd, while deductions made from tests carried out under what are called "practical conditions" by persons who do not understand the real nature of the problem involved are almost always entirely absurd.

Types of Tests.

It is evident, therefore, that tests must be recognised as being broadly divided into two or perhaps three categories.

(1) The scientific test, made with the object of determining coefficients and laws relating to the effect of alteration in conditions.

(2) The rough practical trial, made, for instance, with the object of satisfying a client or demonstrating the accomplishment of a guarantee to the architect or other person responsible.

(3) A species of test which is almost intermediate between the above two categories is what may be called the "manufacturer's" test. This is a test which should be made under more or less scientific conditions on a large practical scale by the manufacturer with the object of ascertaining the efficiency of his product, with a view to improving it, or with a view to giving information to himself or his customers for their guidance. Such tests, if properly carried out by competent observers, possess great value for the practical man. To be properly carried out they must recognise and be conducted in accordance with the fundamental principles of science, which control all scientific investigation, although in the nature

of things they cannot take into account all the factors which the scientist would endeavour to include.

The essential difficulty in designing such tests is to eliminate conditions which would render the result indeterminate, and concentrate attention on main points rather than on side issues, but nevertheless taking care that such side issues as are disregarded are not of sufficient importance to produce any considerable variation in the result, and to do all this with a comparatively limited amount of plant and scientific instruments, and in a limited time.

It is the last kind of test with which we shall be chiefly concerned. The first is the concern of the engineering physicist and the second generally means nothing at all.

No test of any kind is of value unless it is made by a person familiar with the effect of all the attendant conditions and with the effect of a change in those conditions. It is sometimes easy for a skilled experimenter to obtain almost any results he desires within certain limits from a test of heating apparatus by varying inconspicuous conditions which he knows have an ultimate effect on the result.

In making practical tests on heating plant a skilled scientific experimenter must of necessity shut his eyes to many things which he knows ought not to be disregarded. If an attempt were made to take account of all conditions which might have a greater or smaller influence on the result, the carrying out of a test would be so laborious and would occupy so much time that in practice it would be impossible to carry out.

We shall not in this volume deal with tests under category (1), as these are the province of the engineering physicist and descriptions of such tests demand far more space than is here available.

Tests in section (2) are, properly speaking, not tests at all, but trials. They are the only form of test with which the contractor comes directly into contact.

By far the greater proportion of such trials with which the heating engineer has to do consist of trials of completed plants in a building.

The necessity for these trials arises as follows :—

The engineer, as a condition of his obtaining the order, gives a guarantee that certain temperature results will be secured within the building with a given outside temperature. The essential absurdity of such guarantees has been fully discussed in Chapter XVII.

After the installation is completed he has sometimes to "demonstrate" that the guaranteed results have been accomplished. The object of such guarantees and demonstrations is to satisfy the client or purchaser that he obtains an article which reasonably satisfies his requirements, and those who do not understand the subject, and even many people who ought to understand it, imagine that the fulfilment of this guarantee can be demonstrated by a short test.

The facts in connection with this matter have been already dealt with.

It is evident, therefore, that the conditions of an ordinary guarantee are so loose that they have no precise meaning. In order to be as precise as practically possible, the guarantee should state—

(1) The external and internal temperatures.

(2) The actual amount of air which is to pass through the room per hour.

(3) The velocity and direction of the wind.

(4) The degree of humidity of the atmosphere.

(5) The length of time during which the apparatus has been maintained at full power before the thermometer readings are taken.

(6) The length of the intervals of discontinuity of heating.

(7) In order to provide that the test shall be possible, provision should be made for the inevitable variations in the external temperature. The absurdity of the current conventions on this point has been already commented on (Chapter XVII.).

It must be clearly pointed out that any variation from the absolutely strict terms of the guarantee throw the whole result into a state of uncertainty. Accurate information does not exist at the present time and perhaps cannot exist to enable variations in external temperature, wind velocities, dampness, time of working, and other similar matters to be correctly allowed for.

The scientific engineer knows very well that the temperature in the building will continue to rise for a fortnight or even more after the apparatus is first started. The amount of such rise is difficult or impossible to determine beforehand, and very few, even of the most expert engineers, can give any reliable forecast of what the rate of rise per diem will be. The tester knows from experience that the external temperature will not remain constant during that period. No man can say what is the condition of the walls as regards dryness or dampness, or even what is called conductivity. It is impossible to forecast what the wind and weather will be during the interval.

The demonstration must, however, be arranged to take place on a definite date, and the engineer must take his chance of what the conditions will be at that date.

It will be seen, therefore, that the desired result cannot be certainly demonstrated unless the actual power of the apparatus is largely in excess of the minimum power necessary to fulfil the strict terms of the guarantee.

On Measurements of Heat Losses from a Room.

It has been frequently stated in this book that co-efficients determined on scientific principles are only indirectly of use to the heating engineer, because of the wide difference which exists between the conditions under which such co-efficients are determined and those the practical use of the apparatus involves. This is particularly the case in connection with the calculation of such matters as losses of heat from buildings, without doubt the most difficult, as well as the most common, of all the matters with which the heating engineer deals.

An *accurate* determination of the loss of heat from a room, under given conditions, can only be obtained by experiment on the room itself. It is only in rare cases that it can be *accurately* calculated from the plans and drawings of the room. This can be done, however, when the construction of the building is known with accuracy, not otherwise.

It has been previously pointed out that the heating engineer is concerned not only with loss of heat, but with much more subjective matters, such as the feelings produced in the minds of the people who use the appliance.

When the fundamental features of different methods of heating are analysed one can see that they show so much difference that it is almost impossible to compare them. The difficulty, for instance, of equating the effect of an open fire in a room to that of a radiator in the same room has been already discussed.

A thermometer in a room heated by an open fire may stand at 50°. The same room when heated by a radiator may show a temperature on the same thermometer of 60°, yet the "feeling" in the fire-heated room may be warmer than the radiator-heated room.

The heating engineer must have some instrumental means of comparing them. The methods he employs for comparison are at present of a most inefficient character.

No means have yet been devised of comparing the effect of heating rooms by different methods, and it is open to doubt whether such methods can be discovered. The most that can be done at present is to measure the heat lost when rooms are heated by certain specified methods to a specified degree. That with which the heating engineer is most concerned is certainly the method of heating by radiators.

All such investigations involve necessarily the provision of means (1) for determining with precision the state of a room, and (2) for determining the amount of heat lost from the room when maintained in that state.

The state of a radiator-heated room can be determined as completely as present knowledge allows, by observing (A) the temperature of the air at various points, (*B*) the amount of air which passes through it per hour, and (C) the temperature of the surface of the wall.

(*A*) cannot be determined accurately by thermometers, unless used as described on page 354. An entirely satisfactory method of determining (*B*) is to liberate within the room known quantities of a gas easily recognised by chemical analysis, and by making measurements of the percentage of the gas which exists in the air of the room.

The most suitable gas for this purpose is carbonic acid gas which can be procured fairly cheaply, and if used in the form of liquid the amount of gas liberated can be determined by weighing.

The calculations which have been explained on page 161 will then give the rate of interchange of air which is proceeding in the room at the time the observations are taken.

(C) is determined by thermometers inserted in small holes drilled in the wall.

Perhaps the only entirely satisfactory method of determining the amount of energy lost from a room when heated by a radiator consists in fixing a powerful radiator in the desired position in the room, and supplying it with hot water circulating to it from an apparatus outside the room, which will be shortly described, measuring the temperature of the water on its entry and also on its leaving the room, and measuring also the quantity of water thus flowing per hour, the experiment being continued for such a period that the temperature of the room remains constant under constant conditions for two whole days at least.

The temperature should be maintained in the room for a prolonged period, such that the walls have attained the constant temperature, and in order that the result shall be wholly reliable and wholly satisfactory, the external temperature should be constant during the whole period of the test.

If, as is usual, it is impossible to obtain such a constant temperature for any prolonged period, the test is in so far unsatisfactory, and no accurate deductions can be made from such a test.

The best method of measuring the various quantities during this test consists of the provision of two tanks at different levels outside and away from the room to be heated. An apparatus is provided for heating to any desired degree the water in the upper tank. A pipe connection is made from this to the radiator, and another from the outflow of the radiator to the lower tank, thermometers being fixed in these pipes at the point where they enter the room. The overflow from the lower tank flows into a third tank fixed on the platform of a weighing machine, whereby the accurate weight of water passing through the radiator can be determined. Means are also provided for pumping the water from the lower tank to the upper as occasion requires.

Very valuable results might be obtained by prolonged observations on plant of this character, especially if it were possible to remove the apparatus to various rooms in different situations, but so far as the present writer is aware, no such observations have ever been made. The results that would be obtained from such a series of tests would probably vary according to the position of the radiator in the room, and probably also according to the size of the radiator.

Another method which is less satisfactory than the above, for making the same measurements with a great deal less preparation, consists of burning within the room an accurately measured quantity of gas, oil, or other combustible of known composition and heat value.

The combustible should be burnt approximately in the position occupied by the radiator, and provision for the due distribution of the products of combustion and the observation of the temperature effect should be made. This method has the advantage that the rate of interchange of air can be approximately determined by analysis of the air in the room without any special plant for liberating the known quantity of carbonic acid gas.

Preliminary observations on the quality of gas or other combustible employed may be made to determine with accuracy (1) the amount of heat in B.T.U. per cubic foot or per pound given off by the complete combustion of the gas. (2) The amount of CO_2 given off.

The method of making this test is to increase or diminish, by means of a cock, preferably automatically controlled, the amount of gas burnt according to the indication of a thermometer, so that the reading of the latter is constantly kept at the desired value, say, 60°. This may be accomplished by one of the many thermostats now on the market, which would raise or lower the gas consumption according to requirements.

Accurate observations of the gas consumed at various times would then give with precision the amount of heat necessary to maintain the temperature in various conditions, and also analysis of the room air would give the rate of interchange of air per hour.

Extensive observations of this character would result in the collection of valuable information on many points, which are at present uncertain. This method has been employed by the Author in many experiments with satisfactory results.

It is, however, not entirely satisfactory, since the proportion between the radiated and convected energy cannot be the same as in the case of a radiator. When such results are applied to radiator-heated rooms it leaves open a possibility of large errors. Strictly speaking such results only apply to rooms heated in this manner.

The Method of Determining the Loss of Heat from a Room Heated by Warm Air.

Anemometric observations on the quantity of air passing through a room, together with the difference of inlet and outlet temperature, and exploration of the air temperatures within the room at many different points would give as good results of this method of heating as could be obtained. Such results could not be wholly satisfactory because of the considerable difference in temperature at different levels which this method of heating usually produces.

Investigation of the Heat lost from a Room Warmed by Open Fires.

The chief difficulty in this investigation consists in determining what is the state of such a room as regards temperature.

An open fire heats entirely or largely by radiation pure and simple. The walls are generally supposed to be warmer than the air of the room, since it is they which receive the radiation from the fire and communicate it to the air, although experiments with thermometers do not bear out this theory. Measurement of the air temperature by shielded thermometers, as described in Chapter XV., combined with observations on the radiation by radiometers, would be required for a complete investigation.

It is clear that more heat will be lost from the walls when the room is heated by an open fire than when the air in same room is heated to the same degree (as indicated by a shielded thermometer) by means of a radiator, because under the former conditions the temperature of the walls would be higher.

A fire-heated room will generally be much more freely ventilated than if the same room is heated by a radiator, on account of the powerful upcast which the fire produces in the chimney.

The same method of determining this rate of interchange must be employed as before described, namely, by analysis of the interior air when measured quantities of CO_2 are liberated at regular intervals or uniformly.

The only accurate method of measuring the energy actually communicated to the room by means of an open fire would be to reproduce as far as possible the effect of an open fire under conditions which allow of an accurate measurement being made of the consumption of energy during the test.

In the case of a room heated by an open fire the main portion of the heat is communicated to the room by radiation from the glowing coals. But as explained in the last chapter, the heat obtained from radiation of the glowing coals is not the only heat which finds its way into the room by the fire. The fire warms the grate bars and cheeks of the fire by direct conduction, also the metal work and stone work of the mantelpiece, and the adjacent brickwork, also by conduction.

This obviously communicates a considerable amount of heat to the air in a room in the same manner as does a radiator. They also produce a feeling of warmth within the room which cannot be estimated with accuracy.

There are two ways of measuring the quantity of heat communicated to a room by an open fireplace, the one a positive, and the other a negative one.

The positive method consists of reproducing as accurately as the nature of the case allows, the effect of an open fire by means which are themselves capable of accurate measurement. This method of investigation involves two separate and distinct observations.

Means must be provided (1) for determining objectively the condition in which the room is maintained, so as to ensure that as far as possible the conditions produced by the actual fire are the same, apart from subjective sensations, as those produced by the experimental imitation of it; (2) for measuring the quantity of energy consumed in producing those effects.

The best way of reproducing the effect of an open fire would probably be by means of an electric radiator. The greater part of the energy of an electric radiator is given off in the form of radiation. The defect of the method would be that similar conditions as regards the heating of the brickwork and the mantelpiece, etc., surrounding the actual fire would not be obtained.

It would be necessary either to make some calculated allowances for these differences, or to provide an imitation of this effect by heating these surrounding portions of the brickwork by electrical means to the same degree as is accomplished by an open fire.

Such an investigation could only be made at considerable cost and it is doubtful whether, except as a matter of scientific interest, it would be useful expenditure.

The only practically possible method of carrying out an experimental investigation on these lines, would be to provide a very powerful electric stove in place of the fire, determine the condition of the room by means of thermometers and radiometers, suitably placed, and measure the amount of energy consumed by the electric stove by means of a wattmeter attached to it.

If the thermometers and radiometers placed about the room give similar readings when the electric stove is in operation as when the fire is burning, it would be possible to say with some approach to reasonable accuracy that the readings of the wattmeter would give the amount of heat actually transferred to the room by the fire.

The second or negative method would be to weigh carefully the amount of fuel burnt, and determine its thermal capacity, and to measure the volume and temperature of the flue gases passing away to the atmosphere.

The difference may be supposed to give the amount of heat communicated to the room and walls of the building.

This method has been carried out by various experimenters, but the results obtained have been very inconclusive and unconvincing. In particular the matter was investigated on these lines in the year 1903, in the experiments alluded to in the last chapter. The results of these tests were unsatisfactory, and gave little or no information of any value, except as showing the great difficulty of making tests by this method.

It will thus be evident that very little information exists on these important points.

Test of Loss of Heat from Radiators.

These may be made either with steam or water as a heating medium. The first are considerably easier to carry out, and consist in supplying a radiator with steam at a constant pressure and temperature, and measuring the quantity of water condensed in the radiator, suitable observations being made of the surrounding room temperature.

The results so obtained vary widely from day to day. The only satisfactory method, therefore, is to compare by this means the heat lost from the radiator under test with that lost from a standard radiator which is connected to the same steam pipe, and under exactly the same conditions as the test radiator.

The proportion between the heat lost from two radiators similarly placed is practically constant, and therefore any two radiators may be compared by comparing each of them in turn with the standard radiator.

The chief difficulty in obtaining uniform and accurate results by this method consists in ensuring that the steam admitted to the radiator contains no water, and that all the condensed water withdrawn from the radiator has really been condensed by the radiator itself.

The best method of ensuring this result consists in the super-heating the steam by passing it through a coil of pipes fixed in a gas oven placed in another room. The steam can then be slightly super-heated, thereby ensuring that it carries no water into the radiator. The amount of excess heat so carried by the super-heated steam can be allowed for. This quantity is so small that it may with safety be assumed that the steam actually within the radiator is all in a saturated state, for a very small loss of heat from the steam would reduce its temperature to that corresponding to its pressure.

Difficulty also occurs in ensuring that all the condensed water is promptly drained from the radiator and that neither the radiator nor the steam supplied to it contains air. This is a cause of a good deal of inaccuracy in making these tests.

The absolute heat loss from the radiator is found from the difference between the condensation when the radiator is connected and when the radiator is taken away, and the pipes alone are connected. This eliminates all the loss from the pipes.

The variations in the heat actually given off from a radiator have been discussed in the chapter devoted to that matter.

The most convenient source from which to obtain the steam is to take it from an adjacent steam pipe at a considerably higher pressure than it is proposed to employ in the radiator, say, 20 to 50 pounds per square inch, pass it through a reducing valve, which has, of course, a tendency to dry the steam. The pressure commonly employed for these tests is from 5 to 10 pounds per square inch. The temperature and pressure of the steam on entering the chamber are measured by a thermometer and pressure gauge.

The amount of super-heat which it is desirable to communicate to the steam is about 20°, but if 40° of super-heat are given, the temperature may be maintained much more constant than if the degree of super-heat is lower.

The condensed water is best collected in a moderately large strong cast iron cylindrical vessel provided with a water gauge, and at the lower end with a draw off cock, or steam trap of the float type, which keeps the level of the water inside the chamber constant.

It is desirable that this vessel should be almost full of water, so as to allow as little steam space above the water as possible. The water in the vessel loses much of its heat before it is drawn off.

It is necessary to correct all the results secured by means of a special observation on the steam condensed by all the pipes and fittings, other than the radiator under test. This represents a quantity which, though not absolutely constant in itself, bears a constant ratio to the steam condensed by the radiator.

Tests of Radiators by Water.

These tests are made with plant exactly similar to that which has been already described in determining loss of heat from rooms, namely, by means of two tanks placed at different levels with means for heating the water to any desired degree in the upper one and measuring the quantity of water flowing through to the lower one, and the temperature at inlet and outlet to the radiator.

Investigation of the distribution of the temperature within the radiator is also very desirable and instructive.

It is to be observed that this method of making the test needs no correction for loss of heat from pipes, since the temperatures of the water are taken close to the inlet and outlet of the radiator under test, the difference between them being produced solely by the heat given off from the radiator.

Losses of Heat from Pipes.

These are made by methods precisely similar to those adopted for radiators.

Testing of Non-Conducting Composition.

These tests are directed to obtain figures showing the ratio between the heat lost from a pipe coated with the composition under test and the same pipe uncoated. The method of making the test consists in fixing an uncoated pipe together with a coated one at some distance apart, but both connected to the same source of steam, by means of pipes of similar length and measuring the respective quantities of water condensed in the two pipes.

Constant results on successive days cannot be relied upon, but it is found that the ratio between the steam condensed in two apparatus fixed side by side maintains a fairly constant value. The steam is, of course, super-heated in exactly the same way as is described for radiators.

The length of pipe under test should be as long as practically possible, in order to minimise the effect of the ends. But good results can be obtained from 9 foot lengths of pipe. The flanges or clips with which the ends are covered may be eliminated by making a special observation on the steam condensed in them.

After testing a large number of radiators it is very advisable in the interests of economy of time to make simultaneous tests on five or six or even more similar ranges of pipe.

Observations on the efficiency of coating should also include the determination of the surface temperature which, as has already been explained, affords a very good indication of the quality of the non-conducting composition. These temperatures can be taken by means of ordinary chemical stem thermometers, whose bulbs are placed in actual contact with the surface and covered up by a small lump of cotton wool and clay, or partly fixed, so that while the bulb is in actual contact with the surface it is not in contact with the clay.

Other methods of obtaining the surface temperature are to wrap wire, having a known resistance at various temperatures, round the non-conducting composition, and measuring its electrical resistance when the test is proceeding, by means of a post office box.

Observations on non-conduction composition are of little value unless they last for two or three days.

It is very desirable that the same composition should be applied in two or three different thicknesses, and the results plotted on curves similar to those shown on page 341.

Test of a Complete Heating Plant.

This is at the same time the most common and most difficult of all kinds of apparatus of which to obtain a satisfactory test. This matter has already been discussed in Chapter XVII. The principle should always be borne in mind that the object of making a test is to obtain specific and precise information, and unless the test can be so made that its result does supply such information, it is almost useless to make it at all, except as an informal rough trial.

There are several objects with which a heating installation may be tested. The most usual in practice is the test made with a view to ascertaining whether or not an installation will "maintain" a certain temperature inside a building, corresponding to an assumed outer temperature.

It has already been explained that it is impossible to obtain any constant or satisfactory result from such a test as this except by a prolonged trial of between a week and fortnight's duration, and even in this case the result is indeterminate unless the outside temperature has been fairly constant throughout. It is, of course, no guide whatever to take any temperature readings so long as the temperature is rising, for the result obtained will simply depend on the length of time the test is continued, and any result obtained from such a test cannot be called a test at all. It would only become a test in the strict sense if allowance could be made in the calculation for the time element which cannot be done accurately.

Assuming, therefore, that the temperature to be determined is the ultimate temperature which can be maintained by the apparatus, an approximation to this may be obtained in a shorter time than that required to attain the full temperature by plotting the values of the temperature attained on a time base. From this diagram a more or less reliable indication may be obtained of the value towards which the interior temperature is tending.

The observations to be taken in making such a test as this, in addition to the room and the outside temperatures and other weather conditions, are—

(1) The quality, weight and time of stoking of the fuel.

(2) The temperature of the flow and return water.
(3) The temperature of the flow and return at each of the radiators and coils in the system.

The best way of obtaining the latter measurement is to have a thermometer cup fixed on the flow pipe and return pipe leading to each coil. If these are filled with mercury or some kind of heavy non-volatile oil, or even water, it is easy to take the temperature of the flow and return water with a thermometer in a short time. Failing this provision in the apparatus an approximate idea of the radiator temperature can be obtained by running water off the air cock on to a thermometer.

The temperature, however, so obtained is not a reliable indication of the mean temperature of the radiator, for reasons which have been already explained. The temperature of the radiator is intended to be a guide to the heat emission, and since the heat emission has been experimentally determined with reference to the temperatures of the flow and return pipes, it is clear that any other temperature than these will only give an indication of the heat emission of a radiator when the relation between the flow temperature, the return temperature, and the actually observed temperature is known.

It has been explained that this relation depends on the method of connection. The most that can be done, therefore, in the reliable determination of the internal temperatures is to plot the observed results on diagrams and deduce from them what information is possible.

Observation of Fuel Consumption.

No test of an installation is complete unless it comprises such observations as will enable, at any rate, an approximate balance sheet of fuel consumption to be obtained. The heat derived from the fuel per pound is divided between the heat delivered to the rooms, heat lost from the surfaces of the pipes, etc., outside the rooms and from the exposed surfaces of the boiler, etc., the heat rejected in the flue gases.

Observations on all these points are an essential in any complete and satisfactory test.

Test of Circulation.

A test which is as fully important from a technical point of view as any of these, is a circulation test, which comprises observations of the time of lighting of the fire and the temperature attained in the flow pipe and return pipe of each radiator. The perfection of the circulation is gauged by the simultaneous rises in temperature at all different points; if the pipe design is correct each radiator should rise in temperature at approximately the same rate. It is not, of course, possible to obtain absolute uniformity, but approximate uniformity only is sufficient.

The practical method of obtaining this result is to start with the apparatus cold and weigh in a certain quantity of fuel into the furnace, light the fire, noting the time, and make observations of the temperature at different points of the installation, noting in each case the time at which the observation was taken.

After the observations are complete curves of temperature should be drawn showing the rises in temperature of the water at all the various points at which observations are being taken. If these curves all lie on the diagram within a moderate distance of one another, the circulation may be known to be uniform; if not the reason should be carefully investigated.

In an accurately designed installation the temperature in all the flow connections to the radiators does not vary more than 5°, and the return pipe temperatures only vary slightly more than the flow pipe. If a flow pipe keeps hot and the return pipe remains comparatively cold, defective circulation to the radiator is proved, probably due to a short circuit, and steps should be taken to investigate the cause.

It is difficult to determine the junction which is responsible for the short circuit otherwise than by referring to an isometric projection of the entire plant.

In very widely extended installations the temperature differences are of course greater than in a smaller one.

Investigation of the Quantity of Water Flowing.

It is always possible to obtain an approximate idea of the amount of water flowing in each branch by investigating the temperature at the beginning and end of the branch by accurate thermometers. The information provided in Chapter XIII. is sufficient to enable the estimation of quantities to be made.

The accuracy of the estimate depends on the accuracy of the value of K and of the thermometric observations. A very small error in the observation of the temperature will cause a considerable difference in the calculation of flow of water, so that this method of investigation is not only somewhat tedious, but cannot be expected to be very accurate. A moderate degree of accuracy can however be secured, although the investigation takes a considerable time. The time is shortened very much by the provision of thermometer cups at various points in the installation.

If these are not provided the only means available for obtaining the temperature of the water is to lay the thermometers on the surface of the pipe, taking care that the bulb is in contact with the pipe, covering the bulb with cotton wool and clay or putty, in such a way that the clay or putty does not touch the bulb of the thermometer.

This method of determining temperatures cannot be regarded as satisfactory inasmuch as the importance of accurate determination is considerable, and as this method cannot be relied upon within one or two degrees. It is, however, so far as the Author is aware, the only method of investigating the actual flow of water in the interior of the pipes in a working installation.

Test of Boilers.

The method adopted by the best firms of boiler makers in testing the output of heat from the boilers is to provide a testing station in which the boiler is mounted on the platform of a large weighing machine, connections are made between the boiler and the smoke flue by means of mercury sealed sleeves, in which the main flue is not supported on the weighing machine, but terminates in an inverted cup, the rim of which dips into an annular trough of mercury, so that the smoke joint is always made, although no stress can be transmitted between the fixed smoke pipe and the boiler flue outlet.

The weighing machine can therefore be operated without any interference. Similar joints are made to the flow and return pipes. These connect to opposite ends of a large coil contained in a tank of water at a higher level than the boiler, through which the hot water from the boiler circulates. The water in the tank is kept cool by an inflow of cold water, and the overflow from the tank is caught in a tank on a second weighing machine and weighed, the temperature of the inflowing and outflowing water is carefully recorded and the B.T.U. delivered by the coil thus measured.

Another method of observing the heat delivered to the water is indicated in Fig. 141, taken from a paper read by the Author before the Institution of Heating and Ventilating Engineers in 1909.

Care is taken that the difference of temperature between the boiler flow pipe and the return pipe shall be about the same as in a working boiler, namely, from 30—50 degrees. The weight on the platform of the weighing machine is then recorded at intervals of perhaps about 10 minutes throughout a test, the difference from time to time representing the amount of fuel that has been burnt in the interval. The draught is generally provided for by a fan.

Observations on the smoke temperature are carefully kept and in some cases the smoke itself is drawn through a coil in another tank of water, so that direct observations are made of the amount of heat retained in the flue gases, which is commonly rejected into the chimney.

There are difficulties in making anemometric observations of the amount of air used, although in the opinion of the present writer these difficulties should not be insuperable.

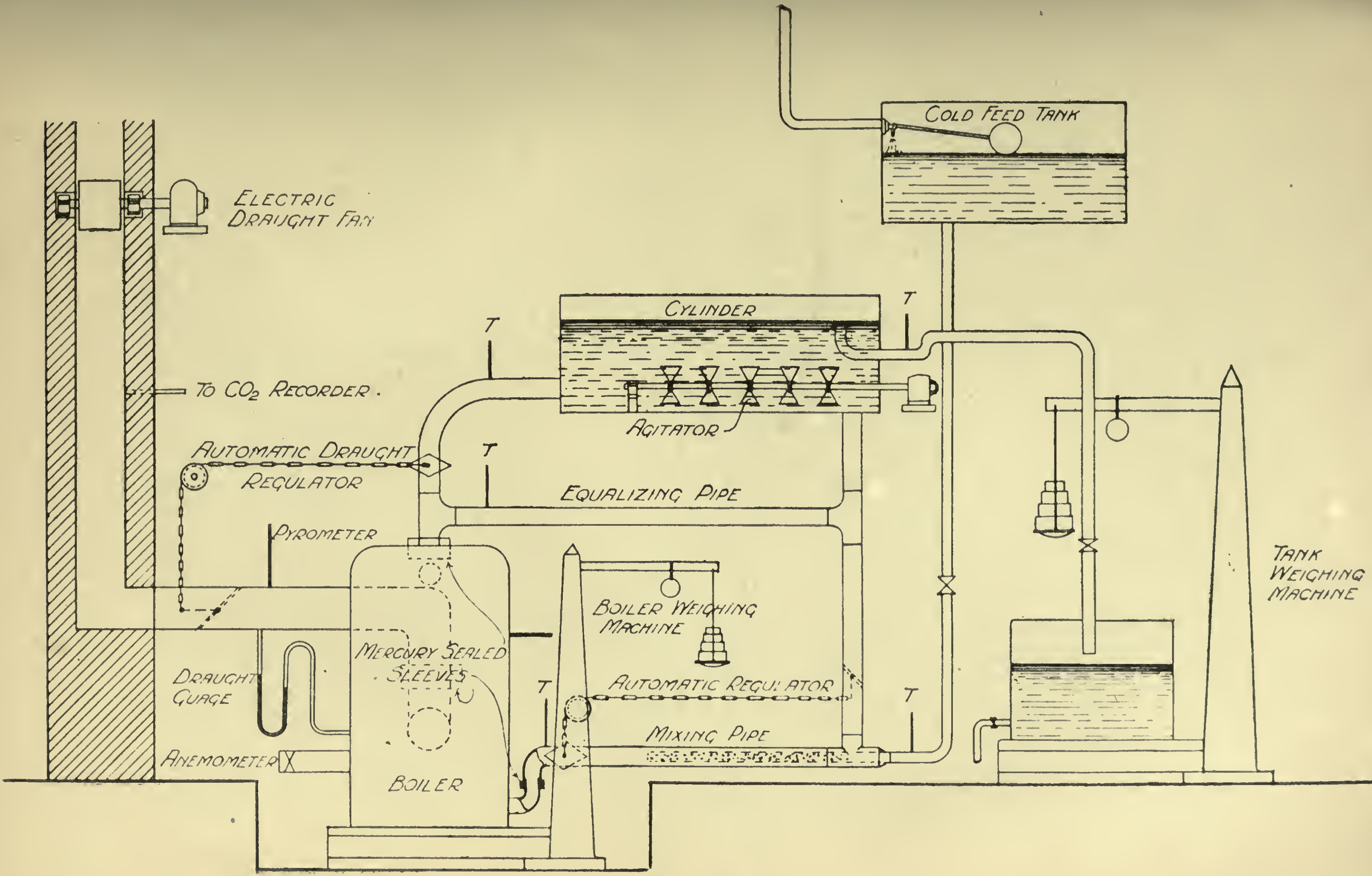

Cold Feed Tank
Electric Draught Fan
Cylinder
T
T
To CO₂ Recorder.
Agitator
Automatic Draught Regulator
T
Equalizing Pipe
Pyrometer
Tank Weighing Machine
Boiler Weighing Machine
Mercury Sealed Sleeves
Draught Guage
T
Automatic Regulator
T
Mixing Pipe
Anemometer
Boiler

FIG. 141.

Periodical analyses of the flue gases are also made. The principle on which they are made is that a continual stream of the flue gases, taken at the boiler outlet, is conducted through a pipe to measuring burettes in which the CO_2, the CO, the oxygen, and the SO_2 are continually recorded for each analysis.

The fuel used is also analysed.

By means of an apparatus of this character very complete information can be obtained about the performance of any boiler. In such a plant of the best class the temperatures are read by means of distant thermometers, so that the whole series can be rapidly taken by an observer at the desk with a minimum waste of time.

Flow of Water in Pipes.

The best method in the opinion of the Author for determining these co-efficients of friction consists of the provision of five tanks of which the highest is a large reservoir of water, the temperature of which can be raised by means of a boiler or steam coil. From this tank the water flows into the second tank in which the level is kept absolutely constant by means of an overflow pipe. From this tank the water flows into the pipe to be tested provided with valves at each end.

Every few feet along this pipe an accurate water gauge is fixed, so that the pressure of water at the corresponding point can be easily determined. At the other end of the test pipe the water flows into a third tank at a lower level than the second, in which the water level can be maintained constant, and at the same time can be adjusted when occasion requires.

The overflow from this tank falls into tank No. 4 which is fixed on a platform of a weighing machine in which the weight of water flowing can be determined at any moment. The water in this tank can be run off into the lowest tank, which is provided with a pump for raising the water again to the first tank, No. 1, so that the experiment can be continued without loss of water.

The friction of bends and elbows and fittings can be determined by passing all the water through the fitting, and measuring accurately the pressure at inlet and at outlet from the fitting. A plant on this principle has been designed by the Author and erected for use in the laboratories at the University College, London.

Losses of Heat Through Wall Surface.

These can be measured by means of an apparatus designed by the Author for use in the laboratories at University College. This consists of a large frame or chamber, the walls of which are formed of non-conducting composition having glazed surfaces. The chamber is open at one side, the area of which is 10 square feet. This is bolted on to the wall the transmission through which is to be tested, and made as air-tight as possible. Within this chamber an air thermometer is fixed, and a similar air thermometer on the outside of the chamber. The pressures in these two thermometers are balanced against one another by a thread of mercury in a tube connecting the two bulbs.

When the temperature in either of the bulbs exceeds that in the other, the thread is forced along the connecting tube, and when the difference exceeds a certain minute fraction an electrical contact is made with wires fused into the tube by means of a "Dynamic Thermometer," supplied by the Dynamic Thermometer Company, Ltd. This causes a weak current of electricity to flow through a coil fixed in the interior of the chamber, so that the air on the inside of the chamber is warmed.

When the temperature of this air again is restored to the same value as that of the room, the dynamic thermometer shuts off the current. By this means the temperature in the interior of the apparatus and of the surrounding air is kept absolutely identical and there is no flow of heat through the non-conducting walls of the chamber.

Any heat that is lost, therefore, from the interior of the chamber must be lost through the surface of the wall. An electric meter records on a strip of paper the amount of electrical energy actually employed in maintaining the temperature, which therefore gives a measure of the total quantity of heat transmitted through 10 feet of the wall surface under test.

Observations are also kept of the temperature in the wall by means of holes drilled into the wall and long chemical stem thermometers.

The Determination of CO_2 in Air.

Fig. 142 shows a most useful and compact apparatus for determining the proportion of CO_2 in atmospheric air, and if carefully used is capable of a considerable degree of accuracy, the limit being probably one half part of CO_2 in 10,000 cubic feet of air.

The principle on which the apparatus works will be understood from a careful inspection of the drawing.

The air burette A, enclosed in a water jacket with a glass face, consists of a wide ungraduated and a very narrow graduated portion. It holds about 20 C.C. from the top to the bottom of the scale. The graduated part, which is 4″ long, is divided into about 100 divisions, each of which corresponds to one 10,000th part of the capacity of the burette when moist. The lowest division is marked zero. Any difference between a reading at or near zero and a second reading is thus shown by the scale in volume per 10,000.

Dr. J. S. Haldane's Apparatus

For Determining the Amount of Carbonic Acid per 10.000 Volumes of Air

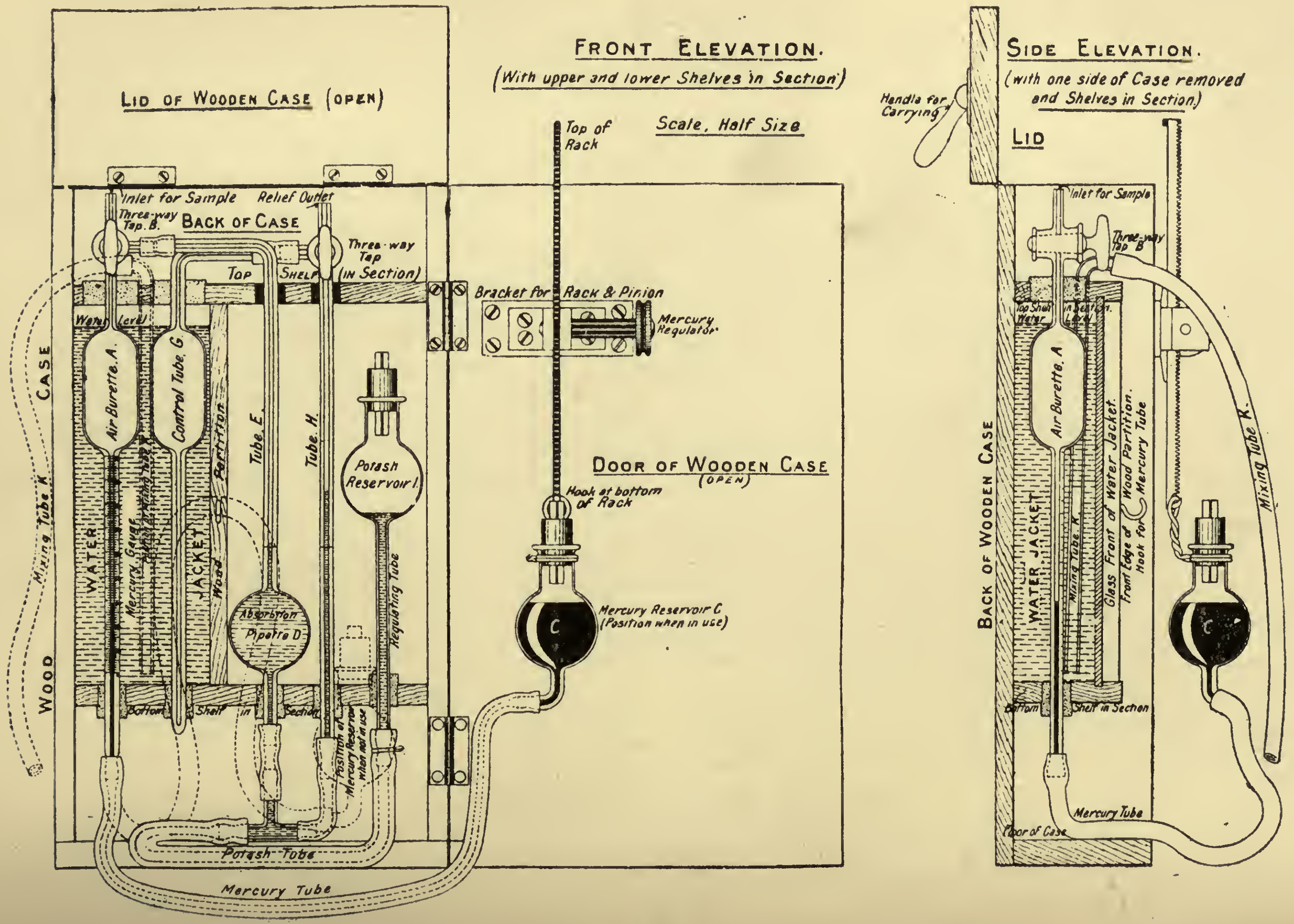


FIG. 142.

In using the apparatus the air is first expelled from the burette by opening the three way tap B to the atmosphere and raising the mercury bulb C. The effect of this is that the mercury rises in the burette A and expels the whole of the air. The bulb is then lowered and placed on the hook of the rack and pinion arrangement F. This causes the mercury in the burette to fall to near the zero mark and draws into the burette a sample of the air to be analysed.

The method of analysis consists in principle in displacing this air in the burette into the absorption pipette D, which contains a solution of caustic potash or soda (about 10%) which has the chemical property of absorbing all the CO_2 which comes in contact with it. After the removal of CO_2 the sample of air is again driven back into the air burette A.

The alteration in volume, owing to the absorption of the CO_2, shows the amount of CO_2 which was originally in the sample of air, that is, the difference between the two readings gives directly the number of volumes of CO_2 per 10,000 in the sample.

The correctness of the analysis depends on the avoidance of errors of various kinds in the two determinations of the volume, owing to variations of pressure or temperature or moisture in the sample of air.

For instance, a variation of about one-fifth of a degree in the temperature in the jacket would, unless compensated, cause an error of 3 volumes per 10,000 in the analysis.

The various methods of avoiding such errors are fully explained in the instructions supplied with the instrument, for which space cannot be spared in this work.

APPENDIX.

TABLE I.

TABLE OF UNITS (see p. 12).

British.		C.G.S.
LENGTH, ETC.—		
1 foot	=	304.8 millimetres
1 inch	=	25.40 millimetres
.0394 inch	=	1 millimetre
39.4 inches	=	1,000 millimetres
3.281 feet	=	1 metre
1 mile per hour	=	.447 metres per sec.
2.236 miles per hour	=	1 metre per sec.
AREA—		
1 sq. in.	=	6.452 sq. cm.
1 sq. ft.	=	929.03 sq. cm. = .0929 sq. metres
10.76 sq. ft.	=	1 sq. metre
0.155 sq. in.	=	1 sq. cm.
VOLUME—		
1 cub. ft.	=	.0283 m^3 = 28.315 litres
1 gallon = 277 cubic ins.	=	4.546 litres
1.76 pint = .22 gallon = 61 cub. in.	=	1 litre
WEIGHT—		
1 lb.	=	0.4536 kilogram = 453.6 grammes.
2.205 lb.	=	1 kilogram = 1,000 gms.
15.43 grains	=	1 gramme
HEAT—		
1° F.	=	$^5/_9$° C. = 0.556° C.
1.8° F.	=	1° C.
1 B.T.U.	=	0.252 calorie
3.968 B.T.U.	=	1 calorie
$\frac{1\ \text{B.T.U.}}{\text{sq. ft.}}$	=	2.71 cal. per sq. metre
$\frac{0.37\ \text{B.T.U.}}{\text{sq. ft.}}$	=	1 cal. per sq. metre
1 B.T.U. per deg. F. per sq. ft.	=	4.88 cal. per deg. C. per sq. metre
0.205 B.T.U. per deg. F. per sq. ft.	=	1 cal. per deg. C. per sq. metre
PRESSURES, ETC.—		
2.04″ mercury = 1 lb. per sq. in.	=	5.17 cm. of mercury = .0703 kilogram per sq. cm. = 70.3 kg. per sq. metre
1″ mercury = 0.49 lb. sq. in.	=	2.54 cm. of mercury = .0344 kilogram per sq. cm.
$\frac{14.223\ \text{lb.}}{\text{sq. in.}}$	=	1 kilogram per sq. cm.
1 foot-pound	=	0.1382 kilogrammetre
7.233 foot-pounds	=	1 kilogrammetre
.205 lb. per sq. ft.	=	1 kilogram per sq. metre = 1 mm. water column

TABLE I. *(Continued)*.

Method of conversion as follows :—

$$1 \text{ calorie per deg. C. per sq. metre} = \frac{1 \text{ cal.}}{1^\circ \text{C.} \times 1 \text{ m}^2} = \frac{3.968 \text{ B.T.U.}}{1.8^\circ \text{F.} \times 10.76 \text{ sq. ft.}} = 0.205 \frac{\text{B.T.U.}}{^\circ\text{F.} \times \text{sq. ft.}}$$

SUNDRY ANTIQUE UNITS OCCASIONALLY USED.

1 gallon = 4 quarts = 8 pints = 277 cub. ins. = 4.544 litres.

36 gallons = 1 barrel.

1 pound (avoirdupois or ordinary pound) = 16 ounces = 256 drachms = 7,000 grains.

1 pound (" troy weight ") = 5,760 grains (same " grains " as above).

1 hundredweight = 112 pounds.

" *Chaldron.*"—There appears to be no exact information as to the meaning of this unit. Originally a measure of volume or cubic capacity, it has apparently a different value in different districts.

Under " Coal and Coke Measure " 1 chaldron = 58.66 cubic feet = 36 bushels = 12 sacks and 4 " pecks " or 8 gallons = 1.284 cubic feet = 2,218 cubic inches = 1 bushel.

Under " Coal Weight " 88 lbs. = 1 bushel, 26½ cwt. = 1 chaldron (London), 53 cwt. = 1 chaldron (Newcastle), generally 1 chaldron of coke = about 12 cwt.

TABLE II.

PROPERTIES OF VARIOUS SUBSTANCES.

Nature of Substances.	Weight of 1 cubic foot.	Specific Gravity Water = 1.	Specific heat.	Expansion per Degree Fahrenheit Length = 100 feet inches.	Melting Point.	Conductivity in B.T.U. Degrees Fahrenheit and Foot Units.
SOLIDS—*Metals*.						
Platinum	1,342	21.52	.0335	.0057	—	19.8
Gold	1,200	19.245	.0324	.0103	2,156	149
Mercury	849	13.596	.033	—	390	4.1
Lead	712	11.42	.0314	.0190	608	20.3
Silver	655	10.5	.0557	.0127	1,832	272
Copper, sheet	549	8.81	.094	.0115	1,996	203
Bronze, from 84 copper 16 tin ...	534	8.56	—	—	—	75
To 15 copper 85 tin	465	7.45	—	—	—	—
Mean	500	8.06	.094	.0125	1,692	57
Nickel	516	8.28	.109	—	—	—
Brass, 75% cu. 25% zinc, sheet	527	8.45	—	—	—	—
60% cu. 40% zinc, Muntz ...	511	8.20	—	—	—	61
Cast, ordinary	505	8.10	.094	.0126	—	54

Steel	490	7.86	.117	.00763	2,372 2,552	20
Wrought iron, mean	480	7.70	.110	.00814	2,912	40.5
Cast iron, mean	450	7.22	.130	.00740	1,922 2,200	40.5
Tin	462	7.41	.057	.0145	446	35.7
Zinc...	449	7.20	.096	.020	680	71
Aluminium, cast	160	2.56	—	—	—	85
Non-Metals.						
Limestone	168 to 178	2.69 to 2.86	0.217	.0415	—	1.35
Marble	157 to 177	2.52 to 2.84	0.215	.00566	—	1.89
Granite	147 to 190	2.36 to 3.06	0.215	.0200	—	193
Chalk	150 to 159	2.46 to 2.55	0.215	—	—	—
Slate	162 to 177	2.60 to 2.85	—	.013	—	0.196
Sandstone	130 to 157	2.08 to 2.52	—	.0102	—	0.88
Glass, flint	187	3.00	0.19	.00575	—	0.53
Ordinary	152 to 168	2.44 to 2.70	0.19	.00632	—	0.53
Portland cement	78 to 94	1.25 to 1.51	0.2	—	—	0.337
Concrete, from	120	1.92	0.2	—	—	0.405
To	140	2.24	—	—	—	—
Mortar	109	1.75	0.2	—	—	—
Brick	125 to 135	2.00 to 2.17	0.2	.00144	—	—
Brickwork	110 to 115	1.76 to 1.84	0.2	.00144	—	0.465
Masonry, rubble	115 to 143	1.85 to 2.30	0.2	—	—	—
Tiles	125	2.00	0.2	—	—	—
Clay	120	1.92	0.2	—	—	—
Sand	118	1.90	0.2	—	—	0.0182
Coal, anthracite, solid	85 to 100	1.37 to 1.59	0.201	—	—	—
Heaped	53	—	—	—	—	—
Coal, bituminous, solid	75 to 80	1.2 to 1.3	0.241	—	—	—
Heaped	45 to 50	—	—	—	—	—
Coke, solid	57 to 100	0.92 to 1.66	0.203	—	—	3.37
Heaped	—	—	—	—	—	0.108
Ice	57.5	0.922	0.504	—	—	0.55
Peat...	6 to 23 mean 11	— —	— —	— —	— —	— —
Mud	102	—	—	—	—	—

TABLE II. *(Continued).*

Nature of Substances.	Weight of 1 cubic foot.	Specific Gravity Water = 1.	Specific heat.	Expansion per Degree Fahrenheit Length = 100 feet inches.	Melting Point.	Conductivity in B.T.U. Degrees Fahrenheit and Foot Units.
Woods, etc.						
Lignum vitæ	40 to 80	0.65 to 1.30	—	—	—	—
American pine	29	0.46	0.650	—	—	0.118
Pear	45.5	0.73	0.500	—	—	—
Oak...	58	0.93	0.570	—	—	0.142
Walnut	42 to 57	0.68 to 0.92	—	—	—	—
Beech	47 to 53	0.75 to 0.85	—	—	—	—
Mahogany	53	0.85	—	—	—	—
Charcoal, solid	90 to 97	0.2 to 0.25	0.242	—	—	0.405
Heaped	13 to 15	—	—	—	—	—
Wool	100.4	1.61	—	—	—	0.216
Human body	67	1.07	—	—	—	—
Cotton	121	1.95	—	—	—	0.027
Gutta percha	60.5	0.97	—	—	—	—
Cork	—	—	—	—	—	0.175
Paper	—	—	—	—	—	0.023
Liquids.						
Mercury	848.7	13.596	0.033	1.015	—	—
Water	62.4	1.00	1.00	1.047	—	0.35
Sea water	64.04	1.026	—	1.05	—	—
Petroleum	54.9	0.88	0.416	1.07	—	—
Sulphuric ether	44.9	0.72	0.66	1.07	—	—
Benzine	53.1	0.85	0.393	—	—	—
Tar	62.4	1.00	—	—	—	—
Oil, olive	—	—	0.309	1.08	—	0.33
Gases at 32° F.		air = 1				
Air	0.0807	1.00	0.238	1.366		0.027
Oxygen	0.0893	1.106	0.218	1.366	—	—
Nitrogen	0.0786	0.974	0.244	1.367	—	—
Carbonic acid	0.123	1.529	0.216	1.369	—	—
Carbonic oxide	0.0781	0.967	0.248	1.367	—	—
Coal gas	0.0354	0.438	0.59	1.368	—	—
Hydrogen	0.00559	0.069	3.405	1.367	—	—

TABLE II.*a* (see p. 244).

TABLE OF ABSOLUTE CONDUCTIVITIES *K*.

$$W \text{ (B.T.U.)} = \frac{A \text{ (ft.}^2\text{)}\ K\ (t_1 - t_2)^\circ\ T \text{ (hours)}}{E \text{ (feet)}}$$

	From	To	Mean for general use.
METALS.			
Copper	176	270	203
Aluminium	83.5	88	85
Zinc	62	71	71
Brass	48.7	73	54
Iron (cast or wrought)	34	49	40.5
Steel	15	34	20
Mercury	4.1	4.75	4.1
BUILDING MATERIALS.			
Marble (fine grain)	—	—	2.4
,, (coarse)	—	—	1.9
Limestone	1.15	1.42	1.35
Freestone	0.88	1.42	1.42
Sandstone	—	—	0.9
Brickwork	0.47	0.48	0.47
Glass	0.50	0.60	0.55
Plaster of Paris	0.23	0.43	0.33
Slate	—	—	0.2
Oak wood (in direction of fibres)	—	—	0.142
Fir wood (parallel to fibre)	—	—	0.12
Fir wood (perp.)	—	—	0.07
Leather	—	—	0.10
Roof paper	—	—	0.08
NON-CONDUCTING MATERIALS.			
Coke (solid)	—	—	3.4
Sand	—	—	0.018
Cork	—	—	0.017
Coke (broken up)	—	—	0.11
Kieselguhr comp.	—	—	0.092
Chalk dust	—	—	0.061
Pumice stone	—	—	0.056
Wood ashes	—	—	0.054
Sawdust	—	—	0.031
Paper	—	—	0.029
Wool	—	—	0.021
Felt	—	—	0.022
Air (at rest)	—	—	0.027
Cotton wool	—	—	0.027
SUNDRIES.			
Water (at rest)	—	—	0.35
Oil (olive)	—	—	0.33
Alcohol	—	—	0.12
Glycerine	—	—	0.16
Ice	—	—	0.55

TABLE III. (see p. 24).

TABLE OF EQUIVALENT TEMPERATURES.

CENTIGRADE AND FAHRENHEIT.

$(C. \times \frac{9}{5}) + 32 = F.$

Degrees Cent.	Degrees Fahr.	Degrees Cent.	Degrees Fahr.	Degrees Cent.	Degrees Fahr.	Degrees Cent.	Degrees Fahr.	Degrees Cent.	Degrees Fahr.
— 20	— 4.0	+ 11	+ 51.8	+ 42	107.6	+ 73	+ 163.4	+ 104	+ 219.2
19	2.2	12	53.6	43	109.4	74	165.2	105	221.0
18	0.4	13	55.4	44	111.2	75	167.0	106	222.8
17	+ 1.4	14	57.2	45	113.0	76	168.8	107	224.6
16	3.2	15	59.0	46	114.8	77	170.6	108	226.4
15	5.0	16	60.8	47	116.6	78	172.4	109	228.2
14	6.8	17	62.6	48	118.4	79	174.2	110	230.0
13	8.6	18	64.4	49	120.2	80	176.0	111	231.8
12	10.4	19	66.2	50	122.0	81	177.8	112	233.6
11	12.2	20	68.0	51	123.8	82	179.6	113	235.4
10	14.0	21	69.8	52	125.6	83	181.4	114	237.2
9	15.8	22	71.6	53	127.4	84	183.2	115	239.0
8	17.6	23	73.4	54	129.2	85	185.0	116	240.8
7	19.4	24	75.2	55	131.0	86	186.8		
6	21.2	25	77.0	56	132.8	87	188.6		
5	23.0	26	78.8	57	134.6	88	190.4		
4	24.8	27	80.6	58	136.4	89	192.2		
3	26.6	28	82.4	59	138.2	90	194.0		
2	28.4	29	84.2	60	140.0	91	195.8		
1	30.2	30	86.0	61	141.8	92	197.6		
0	32.0	31	87.8	62	143.6	93	199.4		
+ 1	33.8	32	89.6	63	145.4	94	201.2		
2	35.6	33	91.4	64	147.2	95	203.0		
3	37.4	34	93.2	65	149.0	96	204.8		
4	39.2	35	95.0	66	150.8	97	206.6		
5	41.0	36	96.8	67	152.6	98	208.4		
6	42.8	37	98.6	68	154.4	99	210.2		
7	44.6	38	100.4	69	156.2	100	212.0		
8	46.4	39	102.2	70	158.0	101	213.8		
9	48.2	40	104.0	71	159.8	102	215.6		
10	50.0	41	105.8	72	161.6	103	217.4		

TABLE III.*a* (see p. 24).

TABLE OF EQUIVALENT TEMPERATURES.

FAHRENHEIT AND CENTIGRADE.

$$(F. - 32)\frac{5}{9} = C.$$

Deg. Fahr.	Degrees Centigrade.	Degrees Fahr.	Degrees Centigrade.	Degrees Fahr.	Degrees Centigrade.	Degrees Fahr.	Degrees Centigrade.	Degrees Fahr.	Degrees Centigrade.	Degrees Fahr.	Degrees Centigrade.	Degrees Fahr.	Degrees Centigrade.	Degrees Fahr.	Degrees Centigrade.
0	—17.78	+31	0.56	+62	16.67	+93	+33.89	+124	+51.11	+155	+68.34	+186	+85.56	+217	+102.78
+1	—17.23	32	0.00	63	17.23	94	34.45	125	51.67	156	68.90	187	86.11	218	103.34
2	16.67	33	+0.56	64	17.78	95	35.00	126	52.23	157	69.45	188	86.67	219	103.90
3	16.11	34	1.11	65	18.34	96	35.56	127	52.78	158	70.00	189	87.23	220	104.45
4	15.56	35	1.67	66	18.89	97	36.11	128	53.34	159	70.56	190	87.78	221	105.00
5	15.00	36	2.23	67	19.45	98	36.67	129	53.90	160	71.11	191	88.34	222	105.56
6	14.45	37	2.78	68	20.00	99	37.23	130	54.45	161	71.67	192	88.90	223	106.11
7	13.90	+38	+3.34	69	20.56	100	37.78	131	55.00	162	72.23	193	89.45	224	106.67
8	13.34	39	3.90	70	21.11	101	38.34	132	55.56	163	72.78	194	90.00	225	107.23
9	12.78	40	4.45	71	21.67	102	38.90	133	56.11	164	73.34	195	90.56	226	107.78
10	12.23	41	5.00	72	22.23	103	39.45	134	56.67	165	73.90	196	91.11	227	108.83
11	11.67	42	5.56	73	22.78	104	40.00	135	57.23	166	74.45	197	91.67	228	108.90
12	11.11	43	6.11	74	23.34	105	40.56	136	57.78	167	75.00	198	92.23	229	109.45
13	10.56	44	6.67	75	23.90	106	41.11	137	58.34	168	75.56	199	92.78	230	110.00
14	10.00	45	7.23	76	24.45	107	41.67	138	58.90	169	76.11	200	93.34	231	110.56
15	9.45	46	7.78	77	25.00	108	42.23	139	59.45	170	76.67	201	93.90	232	111.11
16	8.89	47	8.34	78	25.56	109	42.78	140	60.00	171	77.23	202	94.45	233	111.67
17	8.34	48	8.89	79	26.12	110	43.34	141	60.56	172	77.78	203	95.00	234	112.23
18	7.78	49	9.45	80	26.67	111	43.90	142	61.11	173	78.34	204	95.56	235	112.78
19	7.23	50	10.00	81	27.23	112	44.45	143	61.67	174	78.90	205	96.11	236	113.34
20	6.67	51	10.56	82	27.78	113	45.00	144	62.23	175	79.45	206	96.27	237	113.90
21	6.11	52	11.11	83	28.34	114	45.56	145	62.78	176	80.00	207	97.23	238	114.45
22	5.56	53	11.67	84	28.89	115	46.11	146	63.34	177	80.56	208	97.78	239	115.00
23	5.00	54	12.23	85	29.45	116	46.67	147	63.90	178	81.11	209	98.34	240	115.56
24	4.45	55	12.78	86	30.00	117	47.23	148	64.45	179	81.67	210	98.90	250	121.11
25	3.90	56	13.34	87	30.55	118	47.78	149	65.00	180	82.23	211	99.45	260	126.67
26	3.34	57	13.90	88	31.11	119	48.34	150	65.56	181	82.78	212	100.00	280	137.78
27	2.78	58	14.45	89	31.67	120	48.90	151	66.11	182	83.34	213	100.56	300	148.90
28	2.23	59	15.00	90	32.22	121	49.45	152	66.67	183	83.90	214	101.11	350	176.67
29	1.67	60	15.56	91	32.78	122	50.00	153	67.23	184	84.45	215	101.67	400	204.45
30	1.11	61	16.11	92	33.33	123	50.56	154	67.78	185	85.00	216	102.23	500	260.0

TABLE IV. (see Fig. 3*a*).

DENSITY OF WATER AT VARIOUS TEMPERATURES.

Temperature.	Density lbs. per cubic foot.	Cubic inches per lb.	Temperature.	Density.	Cubic inches per lb.	Temperature.	Density.	Cubic inches per lb.	Temperature.	Density.	Cubic inches per lb.	Temperature.	Density.	Cubic inches per lb.
32	62.418	27.684	112	61.860		140	61.388	28.123	168	60.827		196	60.180	
35	62.422		113	61.844		141	61.370		169	60.805		197	60.154	
39.1	62.425		114	61.829		142	61.351		170	60.783	28.429	198	60.130	
40	62.425	27.681	115	61.813		143	61.331		171	60.762		199	60.105	
45	62.422		116	61.798		144	61.312		172	60.741		200	60.081	28.761
50	62.409	27.688	117	61.781		145	61.291		173	60.720		201	60.055	
55	62.394		118	61.766		146	61.274		174	60.698		202	60.030	
60	62.372	27.705	119	61.750		147	61.254		175	60.673		203	60.003	28.8
62	62.360		120	61.734	27.991	148	61.237		176	60.652		204	59.978	
65	62.344		121	61.717		149	61.218		177	60.630		205	59.950	
70	62.313	27.731	122	61.701		150	61.201	28.235	178	60.606		206	59.925	
75	62.275		123	61.685		151	61.180		179	60.583		207	59.900	
80	62.232	27.767	124	61.670		152	61.159		180	60.560	28.533	208	59.873	
85	62.182		125	61.655		153	61.138		181	60.585		209	59.847	
90	62.133	27.811	126	61.639		154	61.118		182	60.511		210	59.820	28.887
95	62.084		127	61.622		155	61.097		183	60.487		211	59.792	
100	62.031	27.851	128	61.605		156	61.077		184	60.464		212	59.769	
101	62.017		129	61.588		157	61.057		185	60.441		230	59.36	29.110
102	62.002	27.87	130	61.571	28.065	158	61.038		186	60.416		250	58.75	
103	61.989		131	61.552		159	61.018		187	60.393		270	58.18	
104	61.975		132	61.535		160	60.998	28.328	188	60.370		290	57.59	30.005
105	61.960		133	61.517		161	60.978		189	60.347		298	57.27	
106	61.946		134	61.499		162	60.955		190	60.324	28.645	338	56.14	
107	61.932		135	61.481		163	60.935		191	60.300		366	55.29	
108	61.918		136	61.462		164	60.914		192	60.276		390	54.54	31.112
109	61.903		137	61.444		165	60.893		193	60.251				
110	61.890	27.921	138	61.426		166	60.871		194	60.228				
111	61.874		139	61.407		167	60.850		195	60.203				

TABLE IV.*a* (see Fig. 3*b*).

TABLE OF PROPERTIES OF SATURATED STEAM.

Absolute pressure per sq. in. lbs.	Temperature or boiling point. Deg. F.	Total heat in Fah. deg. from water at 32 deg. Fah. Fahr.	Latent heat deg. Fah. Fahr.	Weight 1 cub. ft. Steam. lbs.	Volume 1 lb. Steam. Cub. ft.	Cub. ft. of steam from 1 cub. ft. of water at 62 deg. Fah. Cub. ft.
1	102.1	1112.5	1042.9	.0030	330.36	20,600
2	126.3	1119.7	1025.8	.0058	172.08	10,730
3	141.6	1124.6	1015.0	.0085	117.52	7,327
4	153.1	1128.1	1006.8	.0112	89.62	5,589
5	162.3	1130.9	1000.3	.0138	72.66	4,530
6	170.2	1133.3	994.7	.0163	61.21	3,816
7	176.9	1135.3	990.0	.0189	52.94	3,301
8	182.9	1137.2	985.7	.0214	46.69	2,911
9	188.3	1138.8	981.9	.0239	41.79	2,606
10	193.3	1140.3	978.4	.0264	37.84	2,360
11	197.8	1141.7	975.2	.0289	34.62	2,157
12	202.0	1143.0	972.2	.0314	31.88	1,988
13	205.9	1144.2	969.4	.0338	29.27	1,844
14	209.6	1145.3	966.8	.0362	27.61	1,721
14.7	212.0	1146.1	965.2	.0380	26.36	1,644
15	213.1	1146.4	964.3	.0387	25.85	1,611
20	228.0	1150.9	953.8	.0507	19.72	1,229
25	240.1	1154.6	945.3	.0625	15.99	996
30	250.4	1157.8	937.9	.0743	13.46	838
35	259.3	1160.5	931.6	.0858	11.65	726
40	267.3	1162.9	926.0	.0974	10.27	640
45	274.4	1165.1	920.9	.1089	9.18	572
50	281	1167.1	916.3	.1202	8.31	518
55	287.1	1169.0	912.0	.1314	7.61	474
60	292.7	1170.7	908.0	.1425	7.01	437
65	298	1172.3	904.2	.1538	6.49	405
70	302.9	1173.8	900.8	.1648	6.07	378
75	307.5	1175.2	897.5	.1759	5.68	353
80	312.0	1176.5	894.3	.1869	5.35	333
85	316.1	1177.9	891.4	.1980	5.05	314
90	320.0	1179.1	888.5	.2089	4.79	298
95	324.1	1180.3	885.8	.2198	4.55	283
100	327.9	1181.4	883.1	.2307	4.33	270
105	331.3	1182.4	880.7	.2414	4.14	257
110	334.6	1183.5	878.3	.2521	3.97	247
115	338.0	1184.5	875.9	.2628	3.80	237

TABLE V. (see Chap. III. and p. 49).

PROPERTIES OF AIR AT 29.92 INCHES MERCURY = 14.7 LB. PER SQUARE INCH (ATMOSPHERIC PRESSURE).

Temp. deg. F.	Dry air.		Saturated mixture of air and vapour.								B.T.U. in 1 lb. vapour from water at 32 deg.	Air required to hold 1 lb. vapour in sat. mixture.		Expansion of 1000 c. ft. air when kept saturated.	Relative weights of water/air
	Height of column air 10 ft. high at 32 deg. F.	Weight of 1000 cub. ft. dry air.	Total weight of 1000 c. ft. sat. air.	Weight of vapour in 1000 c. ft.	Weight of dry air in 1000 c. ft.	Pressure of vapour in in. mercury absolute.	Pressure of vapour in lb. per sq. in. absolute.	B.T.U. in 1000 c. ft. air above 32 deg. F.	B.T.U. in 1000 c. ft. vap'r. above 32 deg. F.	Total in mixture.		Weight of dry air.	Volume of dry air 62 deg. F.		
° 32	10	80.7	80.5	0.304	80.2	0.181	.0886	0.0	338	338	1,091	259.0	3,401	1,000	722
35	10.061	80.2	80.0	0.34	79.7	0.204	.100	57	371	428	1,092	234.0	3,080	1,071	775
40	10.163	79.4	79.2	0.41	78.8	0.247	.121	150	448	598	1,094	192.0	2,526	1,092	782
45	10.265	78.7	78.3	0.49	77.9	0.299	.1465	241	536	777	1,095	158.9	2,088	1,105	792
50	10.367	77.9	77.5	0.59	76.9	0.361	.1768	329	647	976	1,097	130.4	1,714	1,118	801
55	10.468	77.2	76.7	0.70	76.0	0.433	.212	415	768	1,183	1,098	108.5	1,426	1,133	810
60	10.57	76.4	75.9	0.82	75.1	0.518	.254	500	901	1,401	1,100	91.6	1,203	1,148	818
65	10.67	75.7	75.1	0.97	74.1	0.617	.302	581	1,068	1,649	1,101	76.4	1,004	1,162	826
70	10.78	75.0	74.3	1.14	73.1	0.733	.359	641	1,256	1,897	1,103	66.0	868	1,177	833
75	10.875	74.3	73.4	1.34	72.1	0.868	.425	737	1,479	2,216	1,104	55.0	723	1,144	840
80	10.98	73.6	72.6	1.56	71.1	1.023	.501	811	1,725	2,536	1,106	45.6	599	1,211	847
85	11.08	72.9	71.8	1.82	70.0	1.203	.590	881	2,016	2,897	1,107	38.4	505	1,226	854
90	11.18	72.3	71.1	2.12	69.0	1.401	.687	951	2,351	3,302	1,109	32.5	427	1,247	863
95	11.28	71.7	70.1	2.45	67.6	1.646	.806	1,013	2,721	3,734	1,110	27.6	363	1,270	872
100	11.38	71.0	69.3	2.83	66.4	1.918	.940	1,073	3,147	4,220	1,112	23.5	308	1,294	881
105	11.48	70.4	68.3	3.25	65.1	2.227	1.092	1,129	3,618	4,747	1,113	20.0	263	1,219	889
110	11.58	69.7	67.4	3.73	63.7	2.577	1.262	1,181	4,158	5,339	1,115	17.1	224	1,344	896

115	11.69	69.1	66.5	4.26	62.2	2.977	1.457	1,227	4,764	5,991	1,117	14.6	192	1,373	903
120	11.79	68.5	65.5	4.88	60.6	3.427	1.680	1,268	5,456	6,724	1,118	12.4	163	1,402	910
125	11.89	67.9	64.5	5.54	59.0	3.930	1.925	1,304	6,201	7,505	1,120	10.7	140	1,439	918
130	11.99	67.3	63.5	6.30	57.2	4.502	2.210	1,332	7,062	8,394	1,121	9.1	118	1,475	926
135	12.10	66.7	62.4	7.14	55.2	5.146	2.520	1,352	8,015	9,367	1,123	7.74	102	1,517	934
140	12.20	66.2	61.3	8.06	53.3	5.858	2.87	1,367	9,060	10,430	1,124	6.61	86.8	1,560	942
145	12.30	65.6	60.2	9.09	51.1	6.655	3.265	1,372	10,230	11,610	1,126	5.62	73.8	1,614	951
150	12.40	65.1	58.9	10.22	48.7	7.546	3.690	1,366	11,520	12,880	1,127	4.77	62.6	1,668	960
155	12.51	64.6	57.6	11.5	46.2	8.536	4.180	1,351	12,920	14,270	1,129	4.03	53.0	1,720	967
160	12.61	64.1	56.8	13.3	43.5	9.628	4.720	1,322	15,070	16,390	1,130	3.26	42.8	1,773	975
165	12.71	63.5	54.9	14.3	40.6	10.843	5.320	1,282	16,210	17,490	1,132	2.83	37.1	1,836	982
170	12.82	63.0	53.4	16.0	37.4	12.18	5.970	1,226	18,150	19,380	1,133	2.33	30.7	1,900	990
175	12.92	62.6	51.8	17.7	34.0	13.66	6.690	1,156	20,130	21,290	1,135	1.92	25.2	1,975	997
180	13.02	62.1	50.1	19.7	30.4	15.27	7.480	1,068	22,390	23,460	1,136	1.54	20.3	2,050	1,005
185	13.12	61.6	48.3	21.8	26.5	17.05	8.35	964	24,820	25,780	1,138	1.22	16.0	2,145	1,013
190	13.22	61.1	46.4	24.1	22.3	19.01	9.32	838	27,470	28,310	1,139	0.93	12.2	2,240	1,021
195	13.32	60.7	44.4	26.6	17.8	21.12	10.34	690	30,370	31,060	1,141	0.67	8.8	2,360	1,029
200	13.42	60.2	42.3	29.3	13.0	23.46	11.50	519	33,510	34,030	1,142	0.44	5.8	2,480	1,037
205	13.53	59.7	40.1	32.3	7.85	26.00	12.71	323	36,900	37,220	1,144	0.24	3.2	2,560	1,045
210	13.63	59.3	37.8	35.4	2.32	28.76	14.09	98	40,580	40,680	1,145	0.065	0.86	2,640	1,055
212	13.67	59.1	36.8	36.8	0.00	29.92	14.65	00	42,210	42,210	1,146	0.000	0.000	2,672	1,058

TABLE V.*a*.

WEIGHT OF AIR AND PRESSURE, ETC., OF VAPOUR ABOVE 212° F.

Temp. Deg. F.	Weight 1000 cub. ft. dry air.	Pressure of vapour in inches mercury absolute.	Pressure in lbs. per sq. in. absolute.	Relative weights $\frac{\text{water}}{\text{air.}}$
220	58.4	35.01	17.16	1,072
230	57.6	42.34	20.75	1,086
240	56.8	50.89	24.92	1,100
250	56.0	60.81	29.80	1,116
260	55.2	72.27	35.40	1,132
270	54.4	85.41	41.80	1,148
280	53.7	100.4	49.20	1,164
290	53.0	117.5	57.60	1,180
300	52.3	136.8	66.90	1,195
320	51.0	183.1	89.90	1,226
340	49.7	241.1	118.2	1,257
360	48.5	312.6	153.3	1,288
380	47.3	399.6	195.7	1,319
400	46.2	504.4	246.8	1,350
450	43.7	858	420.0	1,428
500	41.4	1425	700.0	1,505
550	39.4	2040	1000.0	1,583
600	37.5	—	—	1,660

TABLE VI. (see Chap. VI.).

SHOWING THE AMOUNT OF VITIATION OF AIR AND HEAT, ETC., EMITTED BY OCCUPANTS AND LIGHTS IN A ROOM.

Source of Vitiation.	Surrounding temperatures.	Weight.	B.T.U. given off per hour.	CO_2 given off per hour lbs.	CO_2 given off per hour cubic feet at 32^0 F. S.P.T.	Weight of water vapour per hour lbs.	Pure oxygen consumed cubic feet.
Full grown Man—							
Moderately hard work ...	45°	160	550	0.15	1.21	0.13	
	62°	160	450	0.10	0.81	0.32	
Ditto—							
At rest or asleep	62°	160	380	0.085	0.69	0.046	
Woman—							
At work	65°	135	450	0.10	0.81	0.32	
Child—							
10 years	60°	75	200	0.045	0.365	0.10	
Gas—							
Per cubic foot burnt per hour...	—	0.033	530 to 738	0.07	0.565	0.67	1.0
Bray burner, large, 14ft.^{3}/hr....	—	0.462	10,300	0.98	7.80	0.98	14
Argand burner, 7 ft.^{3}/hr. ...	—	0.231	5,150	0.490	3.95	0.49	7
Incandescent burner, 3ft.^{3}/hr.	—	0.099	2,210	0.21	1.69	0.21	3
Petroleum—							
Per 1lb. burnt per hour ...	—	1	20,000	3.1	2.5	1.12	

TABLE VII. (see Figs. 3*e* and 3*f*).

GLAISHER'S FACTORS.

Multiply the difference between the reading of the wet bulb and dry bulb by the factor shown opposite the dry bulb temperature and deduct the product from the dry bulb temperature, the remainder will give the dew point.

Reading of Dry Bulb Deg. F.	Factor the product of which into the difference between the readings of the Dry and Wet Bulb Thermometer deducted from the former gives the Dew point.	Deg. F.	Factor.	Deg. F.	Factor.
		35	2.60	68	1.79
		36	2.50	69	1.78
		37	2.42	70	1.77
		38	2.36	71	1.76
		39	2.32	72	1.75
		40	2.29	73	1.74
		41	2.26	74	1.73
		42	2.23	75	1.72
10	8.78	43	2.20	76	1.71
11	8.78	44	2.18	77	1.70
12	8.78	45	2.16	78	1.69
13	8.77	46	2.14	79	1.69
14	8.76	47	2.12	80	1.68
15	8.75	48	2.10	81	1.68
16	8.70	49	2.08	82	1.67
17	8.62	50	2.06	83	1.67
18	8.50	51	2.04	84	1.66
19	8.34	52	2.02	85	1.65
20	8.14	53	2.00	86	1.65
21	7.88	54	1.98	87	1.64
22	7.60	55	1.96	88	1.64
23	7.28	56	1.94	89	1.63
24	6.92	57	1.92	90	1.63
25	6.53	58	1.90	91	1.62
26	6.08	59	1.89	92	1.62
27	5.61	60	1.88	93	1.61
28	5.12	61	1.87	94	1.60
29	4.63	62	1.86	95	1.60
30	4.15	63	1.85	96	1.59
31	3.70	64	1.83	97	1.59
32	3.32	65	1.82	98	1.58
33	3.01	66	1.81	99	1.58
34	2.77	67	1.80	100	1.57

TABLE VIII. (see p. 44).

COMPOSITION OF BOILER INCRUSTATIONS FROM DIFFERENT WATERS.

Number.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Source.	Sea water.				Brackish water.	River water.				Wells.		Town supply—Edinburgh.	Spring—Slough.	Pits at surface, Prestonpans.	Quarry—Granton.	An acid water.
							Carlisle.		Thames.	Dunbar.	Slough.					
Pressure in lbs. per sq. in.	5	10	20	25 (?)	...	...	...	60	80	...	...	...	...	...	...	...
$C_a\ CO_3$	2.00	3.44	0.34	0.97	43.65	73.85	75.92	8.20	81.45	32.16	25.62	62.95	50.04	1.22	17.31	2.20
$C_a\ SO_4$	34.00	69.77	72.85	85.53	34.78	3.68	3.16	85.01	1.63	5.64	55.92	20.80	29.76	78.32	53.76	—
$M_g\ (OH_2)$ ...	58.00	22.50	15.83	3.39	4.34	2.56	—	4.36	8.10	—	—	—	—	—	—	—
$M_g\ CO_3$	—	—	—	—	—	—	10.16	—	—	20.04	5.56	7.24	10.84	10.36	18.04	None
N_a salts	—	—	—	—	—	—	0.54	—	—	3.31	—	0.86	0.86	0.64	0.54	—
$N_a\ CI$	Trace	0.99	2.16	2.79	0.56	0.45	—	Trace	Trace	—	0.22	—	—	—	—	—
$F_{e2}, O_3, AI_2\ O_3,$ and $CA_3\ (PO_4)_2$ ...	1.33	1.36	2.40	0.32	3.44	2.96	2.96	0.52	2.44	7.46	5.04	2.48	2.36	4.64	2.88	83.80
$S_i\ O_2$	Trace	0.16	0.80	1.10	7.52	7.66	4.94	1.91	0.57	16.94	5.26	3.76	4.28	3.22	4.36	5.28
Organic matter	4.67	1.78	2.62	Trace	1.55	3.64	0.49	Trace	4.65	7.67	1.04	0.69	0.64	0.88	2.33	8.72
Moisture ...				5.90	4.16	3.20	1.53		0.86	6.78	1.34	1.22	1.22	0.72	0.78	

TABLE VIII.*a*.

The composition of sea water is as follows :—

Sodium Chloride	2.5
Magnesium Sulphate	1.53
Magnesium Chloride	0.33
Magnesium Carbonate Calcium Carbonate	0.02
Calcium Sulphate	0.01
Total Solids	3.4
Water	96.6
	100.0

The density of sea water is 64 lbs. per cubic foot.

TABLE VIII.*b* (see p. 44).

SPECIMEN ANALYSES OF VARIOUS WATERS.

(Figures give grains per gallon according to custom.)

1 gallon = 10 lb. water at 62° F. = 277.27 cub. ins. = 4.54 litres.

1 grain = .000143 lb. = .0684 gramme.

	Town supply, Hartlepool.	Town supply, Barry.	Well Water, Battersea.	Canal Water, Birmingham.	Colliery Pit Water.
Carbonate of lime ...	11.39	18.81	15.50	14.27	—
Sulphate of lime ...	—	—	24.38	13.25	—
Carbonate of magnesia	2.77	5.36	4.85	—	14.45
Sulphate of magnesia ...	7.68	3.57	—	10.96	23.10
Total hardness	21.09°	28.16°	47.73°	32.80°	59.50°
Sulphate of sodium ...	15.37	—	—	16.33	—
Chloride of sodium ...	5.33	4.10	14.33	29.90	325.72
Silica	0.19	0.24	0.25	0.25	0.22
Oxide of iron	0.13	0.18	0.15	0.20	0.13
Nitrate of sodium ...	—	2.97	23.45	2.68	11.90
Nitrate of calcium ...	—	—	5.77	—	—
Chloride of calcium ...	—	—	—	—	25.09

TABLE IX. (see Figs. 8*a* and 8*b* and p. 70).

VELOCITIES OF A BODY DUE TO FALLING VARIOUS HEIGHTS.

Height of fall.		Velocity.	Height of fall.		Velocity.	Height of fall.		Velocity.
feet.	inches.	ft./sec.	feet.	inches.	ft./sec.	feet.	inches.	ft./sec.
.0013	1/64	0.29	0.7	8.4	6.71	14	168	30.03
.0026	1/32	0.41	0.8	9.6	7.18	15	180	31.08
.0039	3/64	0.502	0.9	10.8	7.61	16	192	32.10
.0052	1/16	0.57	1.0	12.0	8.03	17	204	33.09
.0078	3/32	0.71	1.2	14.4	8.79	18	216	34.05
.01	0.12	0.803	1.4	16.8	9.50	19	228	34.98
.0104	1/8	0.82	1.6	19.2	10.15	20	240	35.89
0.0156	3/16	1.005	1.8	21.6	10.77	21	252	36.77
0.02	0.24	1.14	2.0	24.0	11.35	22	264	37.64
0.0208	1/4	1.16	2.25	27.0	12.04	23	276	38.49
0.026	5/16	1.29	2.5	30	12.69	24	288	39.31
0.030	0.36	1.39	2.75	33	13.31	25	300	40.12
0.0313	3/8	1.42	3.0	36	13.90	26	312	40.9
0.0365	7/16	1.53	3.5	42	15.01	27	324	41.6
0.04	0.48	1.61	4.0	48	16.05	28	336	42.4
0.0417	1/2	1.64	4.5	54	17.03	29	348	43.3
0.05	0.60	1.80	5.0	60	17.99	30	360	43.9
0.0521	5/8	1.83	5.5	66	18.82	35	420	47.45
0.06	0.72	1.97	6.0	72	19.66	40	480	50.75
0.0625	3/4	2.01	6.5	78	20.46	45	540	53.83
0.07	0.84	2.12	7.0	84	21.23	50	600	56.74
0.073	7/8	2.17	7.5	90	21.97	55	660	59.5
0.08	0.96	2.27	8.0	96	22.69	60	720	62.1
0.09	1.08	2.41	8.5	102	23.40			
0.1	1.2	2.54	9.0	108	24.07			
0.2	2.4	3.20	9.5	114	24.73			
0.3	3.6	4.40	10.0	120	25.38			
0.4	4.8	5.07	11	132	26.62			
0.5	6.0	5.68	12	144	27.80			
0.6	7.2	6.22	13	156	28.93			

TABLE IX.*a* (see Figs. 8*a* and 8*b*).

HEIGHT OF FALL FOR VARIOUS VELOCITIES AND PRESSURE IN INCHES OF WATER NECESSARY TO GENERATE THE VELOCITY IN AIR AT VARIOUS TEMPERATURES.

$\frac{v^2}{2\,g} \times 12 \times$ specific gravity of air at temperature given.

Velocity t./sec.	Height.		Pressure in ins. water required to maintain flow of air at various temperature.						
	Feet. $\frac{v^2}{2\,g}$	Inches. $\frac{v^2}{2\,g} \times 12$	32°	60°	100°	150°	250°	400°	600°
0.25	0.001	0.012	.0000155	.0000147	.0000136	.0000125	.0000107	.00000888	.00000721
0.50	0.0039	0.0467	.0000607	.0000574	.0000531	.0000488	.0000418	.0000346	.0000281
0.75	0.0087	0.104	.000135	.000128	.000118	.000109	.000931	.0000772	.0000627
1.00	0.016	0.192	.000248	.000235	.000218	.000201	.000172	.000142	.000115
1.25	0.024	0.288	.000372	.000353	.000328	.000300	.000257	.000213	.000172
1.50	0.035	0.42	.000544	.000515	.000478	.000438	.000375	.000311	.000252
1.75	0.048	0.575	.000746	.000705	.000655	.000601	.000515	.000426	.000346
2.0	0.062	0.742	.000961	.000910	.000847	.000775	.000665	.000550	.000446
2.5	0.097	1.16	.00152	.00142	.00132	.00121	.00104	.000861	.000700
3.0	0.140	1.68	.00218	.00206	.00191	.00175	.00150	.00124	.00101
3.5	0.190	2.28	.00295	.00280	.00260	.00238	.00204	.00168	.00137
4.0	0.248	2.97	.00385	.00365	.00339	.00312	.00266	.00221	.00178
4.5	0.314	3.76	.00487	.00461	.00428	.00393	.00337	.00280	.00227
5.0	0.388	4.65	.00603	.00571	.00530	.00486	.00416	.00345	.00280
6.0	0.559	6.71	.00869	.00821	.00762	.00700	.00600	.00497	.00403
7.0	0.761	9.12	.0118	.0112	.0104	.00951	.00818	.00680	.00548
8.0	0.994	11.9	.0154	.0146	.0136	.0124	.0106	.00885	.00717

9.0	1.26	15.1	.0196	.0185	.0172	.0158	.0135	.0112	.00910
10.0	1.55	18.6	.0242	.0228	.0212	.0194	.0166	.0138	.0112
11.0	1.88	22.6	.0292	.0277	.0257	.0236	.0202	.0167	.0135
12.0	2.24	26.9	.0348	.0330	.0306	.0281	.0241	.0199	.0161
13.0	2.62	31.4	.0406	.0385	.0357	.0328	.0281	.0233	.0188
14.0	3.04	36.5	.0471	.0446	.0415	.0381	.0326	.0270	.0218
15.0	3.49	41.8	.0542	.0513	.0477	.0438	.0375	.0310	.0252
16.0	3.98	47.8	.0620	.0585	.0543	.0500	.0427	.0355	.0207
17.0	4.49	53.7	.0700	.0660	.0613	.0564	.0481	.0400	.0324
18.0	5.03	60.5	.0781	.0740	.0685	.0630	.0540	.0447	.0362
19.0	5.61	67.3	.0871	.0825	.0768	.0704	.0601	.0500	.0404
20.0	6.21	74.5	.0966	.0912	.0847	.0780	.0667	.0552	.0448
21.0	6.85	82.0	.106	.101	.0935	.0860	.0735	.0610	.0493
22.0	7.52	90.2	.117	.110	.103	.0944	.0810	.0670	.0542
23.0	8.21	98.5	.1275	.121	.112	.103	.0881	.0731	.0591
24.0	8.94	107.0	.1385	.131	.122	.112	.0960	.0795	.0645
25.0	9.71	116.5	.1505	.143	.132	.121	.104	.0863	.0700
30.0	14.0	168.0	.217	.206	.1905	.175	.150	.124	.101
35.0	19.0	228.0	.295	.280	.260	.238	.204	.169	.137
40	24.9	298.8	.386	.366	.338	.311	.267	.221	.180
45	31.4	376.8	.486	.461	.426	.382	.335	.278	.226
50	38.8	465.6	.601	.571	.527	.485	.416	.344	.280
55	47.0	564.0	.728	.690	.640	.587	.503	.417	.338
60	55.0	660.0	.85	.808	.750	.685	.589	.487	.396

TABLE X. (see Figs. 128 to 131).

FLOW OF WATER IN PIPES.

Giving (I.) deliveries of water in pounds per hour at 160° F. by pipes of various sizes at different velocities.
(II.) Corresponding loss of head in 10ft. run of pipe in inches of water column, according to Weisbach formula.

Velocity inches per sec.	I.—Value of $\frac{v^2}{2g}$ inches.	II.—Value of ρ.	III.—Value of $\frac{\rho\ v^2}{64.4}$	Size of pipe.	¼″	⅜″	½″	¾″	1″	1¼″	1½″	2″	2½″	3″	4″	5″	6″
				Area □″	.0491	.110	.196	.442	.785	1.23	1.77	3.14	4.91	7.07	12.57	19.6	28.27
				Value of $\frac{10}{d}$	40	26.65	20	13.33	10	8	6.66	5	4	3.33	2.5	2	1.67
				Value of $\frac{3{,}600 \times a}{28.33}$	6.23	14	24.9	56.2	99.7	156	225	398	623	898	1,595	2,490	3,590
1	.00129	.0739	.00114	I.	6.23	14	24.9	56.2	99.7	156	225	398	623	898	1,595	2,490	3,590
				II.	.0456	.0304	.0228	.0152	.0114	.00913	.0076	.00571	.00456	.0038	.00285	.00228	.0019
1.2	.0020	.0686	.00164	I.	7.5	16.8	29.9	67.4	120	187	270	478	748	1,076	1,820	2,990	4,310
				II.	.0658	.0438	.0329	.0219	.0164	.01316	.0110	.00823	.00658	.00548	.00411	.00329	.00274
1.4	.0029	.0645	.00196	I.	8.74	19.6	34.8	78.5	140	218	315	558	873	1,255	2,230	3,480	5,020
				II.	.0783	.0522	.0392	.0261	.0196	.0157	.0130	.0098	.00784	.00643	.00490	.00392	.00327
1.6	.0033	.0613	.00243	I.	10	22.4	39.8	89.8	160	250	360	638	998	1,435	2,550	3,980	5,730
				II.	.0971	.0647	.0486	.0323	.0243	.0194	.0162	.0121	.00971	.00810	.00657	.00486	.00405
1.8	.0041	.0587	.00288	I.	11.25	25.2	44.8	101	180	281	405	718	1,123	1,615	2,865	4,480	6,450
				II.	.115	.0770	.0576	.0385	.0288	.0231	.0192	.0144	.0115	.0096	.0072	.0058	.0048
2	.00516	.0564	.00348	I.	12.5	28	49.8	112	200	312	450	798	1,245	1,795	3,180	4,980	7,180
				II.	.139	.0928	.0696	.0464	.0348	.0278	.0232	.0174	.0139	.0116	.0087	.0070	.0058
2.2	.0063	.0545	.00412	I.	13.7	30.8	54.8	123.5	220	343	495	878	1,370	1,975	3,500	5,470	7,900
				II.	.165	.1096	.0824	.0548	.0412	.0330	.0274	.0206	.0165	.0137	.0103	.0082	.0069

2.4	.0077	.0528	.00488	I.	15	33.6	59.8	134.5	240	374	540	958	1,495	2,155	3,820	5,970	8,610
				II.	.195	.130	.0976	.0652	.0488	.0390	.0326	.0244	.0195	.0163	.0122	0098	.0082
2.6	.0082	.0513	.00505	I.	16.2	36.3	64.8	146	260	405	585	1,038	1,620	2,330	4,140	6,470	9,320
				II.	.202	.1348	.101	.0774	.0505	.0404	.0337	.0253	.0202	.0169	.0126	.0101	.0084
2.8	.01	.0499	.00596	I.	17.5	39.2	69.8	157	280	437	630	1,117	1,745	2,510	4,460	6,970	10,050
				II.	.2394	.1598	.1197	.0799	.0598	.0479	.040	.0299	.0239	.0200	.0150	.0120	.0099
3	.0116	.0487	.00677	I.	18.7	42	74.8	168	300	468	675	1,197	1,870	2,690	4,780	7,470	10,750
				II.	.2708	.1800	.1354	.0900	.0677	.0542	.0450	.0339	.0271	.0225	.0169	.0135	.0113
3.2	.013	.0477	.00742	I.	20	44.8	79.8	180	320	499	720	1,276	1,995	2,868	5,100	7,966	11,470
				II.	.2968	.1984	.1484	.0992	.0742	.0594	.0496	.0371	.0297	.0248	.0186	.0148	.0124
3.4	.015	.0467	.00840	I.	21.2	47.5	84.8	191	340	530	765	1,356	2,120	3,046	5,420	8,462	12,190
				II.	.336	.224	.168	.112	.084	.0672	.056	.0420	.0336	.028	.0210	.0168	.014
3.6	.0165	.0457	.00904	I.	22.5	50.3	89.8	202	360	561	810	1,435	2,245	3,224	5,740	8,958	12,910
				II.	.3616	.2416	.1808	.1208	.0904	.0723	.0604	.0452	.0362	.0302	.0226	.0181	.0151
3.8	.0185	.0449	.00996	I.	23.7	53.1	94.7	213	380	592	855	1,515	2,370	3,402	6,060	9,454	13,630
				II.	.398	.266	.199	.133	.0996	.0797	.0664	.0498	.0398	.0332	.0249	.0199	.0166
4	.0207	.0441	.01093	I.	25	56	99.7	224	400	623	900	1,594	2,495	3,580	6,380	9,950	14,350
				II.	.437	.292	.219	.146	.109	.0874	.073	.0546	.0437	.0365	.0273	.0219	.0183
4.2	.0227	.0434	.01182	I.	26.2	58.8	104.6	236	420	654	945	1,674	2,620	3,758	6,698	10,448	15,070
				II.	.473	.316	.236	.157	.118	.0946	.0790	.0591	.0473	.0395	.0296	.0236	.0197
4.4	.0245	.0427	.01255	I.	27.4	61.5	109.5	247	440	686	990	1,753	2,745	3,936	7,016	10,946	15,790
				II.	.502	.334	.251	.167	.126	.1004	.0836	.0627	.0502	.0418	.0314	.0251	.0209
4.6	.0273	.0421	.01375	I.	28.7	64.4	114.5	258	460	717	1,035	1,833	2,870	4,114	7,334	11,444	16,510
				II.	.550	.366	.275	.182	.137	.110	.0916	.0687	.0550	.0458	.0344	.0275	.0229
4.8	.0292	.0415	.01453	I.	30	67.2	119.5	269	480	749	1,080	1,912	2,995	4,292	7,652	11,942	17,230
				II.	.581	.388	.291	.194	.147	.116	.097	.0726	.0581	.0485	.0363	.0291	.0243

TABLE X.—(*Continued*).

Velocity inches per sec.	I.—Value of $\frac{v^2}{2g}$ inches.	II.—Value of ρ.	III.—Value of $\frac{\rho v^2}{64.4}$	Size of pipe.	¼"	⅜"	½"	¾"	1"	1¼"	1½"	2"	2½"	3"	4"	5"	6"
				Area □"	.0491	.110	.196	.442	.785	1.23	1.77	3.14	4.91	7.07	12.57	19.6	28.27
				Value of $\frac{10}{d}$	40	26.65	20	13.33	10	8	6.66	5	4	3.33	2.5	2	1.67
				Value of $\frac{3,600 \times a}{28.33}$	6.23	14	24.9	56.2	99.7	156	225	398	623	898	1,595	2,490	3,590
5	.0323	.0409	.01585	I.	31.2	70	124.5	281	500	780	1,125	1,992	3,120	4,470	7,970	12,440	17,950
				II.	.634	.422	.317	.211	.158	.127	.1056	.0792	.0634	.0528	.0396	.0317	.0265
5.2	.0350	.0404	.01697	I.	32.5	72.8	129.5	292	520	811	1,170	2,071	3,245	4,652	8,288	12,938	18,660
				II.	.679	.453	.339	.226	.1697	.1358	.113	.0848	.0679	.0566	.0424	.0339	.0283
5.4	.0373	.0400	.01790	I.	33.7	75.5	134.3	303	540	842	1,215	2,150	3,370	4,834	8,606	13,436	19,370
				II.	.716	.478	.358	.239	.179	.143	.119	.0895	.0716	.0597	.0447	.0358	.0299
5.6	.040	.0395	.01895	I.	35	78.3	139.3	314	560	873	1,260	2,230	3,495	5,016	8,924	13,934	20,080
				II.	.758	.505	.379	.252	.1895	.1516	.126	.0948	.0758	.0631	.0474	.0379	.0316
5.8	.0432	.0391	.02025	I.	36.2	81.1	144.2	325	580	904	1,305	2,310	3,620	5,198	9,242	14,432	20,790
				II.	.810	.540	.405	.270	.2025	.162	.135	.1013	.0810	.0675	.0506	.0405	.0338
6	.0465	.0387	.02160	I.	37.5	84	149.2	337	600	935	1,350	2,390	3,745	5,380	9,560	14,930	21,500
				II.	.864	.576	.432	.288	.216	.1728	.144	.108	.0864	.0720	.0540	.0432	.0361
6.2	.0492	.0383	.02262	I.	38.7	86.8	154.2	348	620	967	1,395	2,470	3,870	5,560	9,878	15,424	22,220
				II.	.905	.604	.452	.302	.226	.181	.151	.113	.0905	.0755	.0566	.0452	.0377
6.4	.0525	.0379	.02390	I.	40	89.5	159.2	359	640	998	1,440	2,550	3,995	5,740	10,196	15,918	22,940
				II.	.956	.638	.478	.319	.239	.191	.159	.119	.0956	.0797	.0598	.0478	.0399

6.6	.0543	.0376	.0245	I. II.	41.2 .980	92.2 .654	164 .490	370 .327	660 .245	1,029 .196	1,485 .163	2,630 .123	4,120 .098	5,920 .0817	10,514 .0612	16,412 .0490	23,660 .0409
6.8	.0597	.0372	.0266	I. II.	42.4 1.064	95.1 .710	168 .532	381 .355	680 .266	1,060 .213	1,530 .178	2,710 .133	4,245 .1064	6,100 .0888	10,832 .0665	16,906 .0532	24,380 .0443
7	.0634	.0369	.0281	I. II.	43.7 1.124	98 .750	174 .562	393 .375	700 .281	1,091 .225	1,575 .187	2,790 .141	4,360 .112	6,280 .0937	11,150 .0702	17,400 .0562	25,100 .0470
7.2	.0670	.0366	.0294	I. II.	45 1.176	100.8 .784	178 .588	404 .392	720 .294	1,122 .235	1,620 .196	2,870 .147	4,485 .118	6,460 .098	11,468 .0735	17,900 .0588	25,820 .0491
7.4	.071	.0363	.0309	I. II.	46.2 1.236	103.5 .824	184 .618	415 .412	740 .309	1,153 .247	1,665 .206	2,950 .156	4,610 .124	6,640 .103	11,786 .0777	18,400 .0618	26,540 .0516
7.6	.074	.0360	.0319	I. II.	47.4 1.278	106.2 .852	189 .639	427 .426	760 .319	1,184 .256	1,710 .213	3,030 .160	4,735 .128	6,820 .106	12,104 .080	18,900 .064	27,260 .053
7.8	.078	.0357	.0333	I. II.	48.7 1.332	109 .888	194 .666	438 .444	780 .333	1,215 .266	1,755 .222	3,110 .167	4,860 .133	7,000 .111	12,422 .083	19,400 .067	27,980 .056
8	.0827	.0354	.0351	I. II.	49.9 1.404	112 .936	199 .702	449 .468	800 .351	1,247 .281	1,800 .234	3,190 .176	4,986 .141	7,180 .117	12,740 .088	19,900 .070	28,700 .0586
8.2	.087	.0351	.0366	I. II.	51.1 1.464	114.8 .976	204 .732	460 .488	820 .366	1,278 .293	1,845 .244	3,270 .183	5,110 .146	7,360 .122	13,058 .0915	20,400 .0732	29,420 .0611
8.4	.091	.0349	.0381	I. II.	52.4 1.524	117.3 1.016	209 .762	471 .508	840 .381	1,309 .305	1,890 .254	3,350 .191	5,235 .152	7,540 .127	13,376 .095	20,900 .076	30,140 .0636
8.6	.095	.0346	.0394	I. II.	53.6 1.576	120.5 1.052	214 .788	482 .526	860 .394	1,340 .315	1,935 .263	3,430 .197	5,360 .158	7,720 .132	13,694 .0985	21,400 .0788	30,860 .0657
8.8	.10	.0344	.0413	I. II.	54.9 1.652	123 1.104	219 .826	493 .552	880 .413	1,371 .331	1,980 .276	3,510 .207	5,485 .165	7,900 .138	14,012 .1032	21,900 .0826	31,580 .0690
9	.1045	.0342	.0428	I. II.	56.1 1.712	126 1.144	224 .856	505 .572	900 .428	1,402 .342	2,025 .286	3,590 .214	5,610 .171	8,080 .143	14,330 .107	22,400 .0856	32,300 .0716

TABLE X.—(*Continued*).

Velocity inches per sec.	I.—Value of $\frac{v^2}{2g}$ inches.	II.—Value of ρ.	III.—Value of $\frac{\rho v^2}{64.4}$	Size of pipe.	¼″	⅜″	½″	¾″	1″	1¼″	1½″	2″	2½″	3″	4″	5″	6″
				Area □″	.0491	.110	.196	.442	.785	1.23	1.77	3.14	4.91	7.07	12.57	19.6	28.27
				Value of $\frac{10}{d}$	40	26.65	20	13.33	10	8	6.66	5	4	3.33	2.5	2	1.67
				Value of $\frac{3,600 \times a}{28.33}$	6.23	14	24.9	56.2	99.7	156	225	398	623	898	1,595	2,490	3,590
9.2	.109	.0340	.0445	I.	57.4	128.8	229	516	920	1,433	2,070	3,670	5,735	8.260	14,698	22,900	33,020
				II.	1.78	1.188	.890	.594	.445	.356	.297	.223	.178	.149	.111	.089	.0743
9.4	.114	.0337	.0460	I.	58.6	131.3	234	527	940	1,464	2,115	3,740	5,860	8,440	14,966	23,400	33,740
				II.	1.840	1.226	.920	.613	.460	.368	.307	.230	.184	.153	.115	.092	.0769
9.6	.118	.0336	.0475	I.	59.9	134	239	538	960	1,496	2,160	3,830	5,985	8,620	15,284	23,900	34,460
				II.	1.90	1.268	.950	.634	.475	.380	.317	.237	.190	.158	.119	.095	.079
9.8	.124	.0334	.0497	I.	61.1	137	244	550	980	1,528	2,205	3,910	6,110	8,800	15,602	24,400	35,180
				II.	1.988	1.328	.994	.664	.497	.398	.332	.249	.199	.166	.124	.099	.083
10	.129	.0332	.0513	I.	62.3	140	249	562	1,000	1,560	2,250	3,980	6,235	8,980	15,920	24,900	35,900
				II.	2.052	1.368	1.026	.684	.513	.410	.342	.257	.205	.171	.128	.103	.0857
10.2	.134	.0330	.0530	I.	63.7	143	254	573	1,020	1,591	2,295	4,060	6,360	9,160	16,244	25,400	36,620
				II.	2.120	1.416	1.06	.708	.530	.424	.354	.265	.212	.177	.132	.106	.0885
10.4	.139	.0329	.0548	I.	65	145.5	259	584	1,040	1,622	2,340	4,140	6,485	9,340	16,568	25,900	37,340
				II.	2.192	1.464	1.096	.732	.548	.438	.366	.274	.219	.183	.137	.1096	.0915
10.6	.145	.0327	.0568	I.	66.1	148.2	264	595	1,060	1,653	2,385	4,220	6,610	9,520	16,892	26,400	38,060
				II.	2.272	1.516	1.136	.758	.568	.454	.379	.284	.227	.189	.142	.114	.095

10.8	.150	.0325	.0585	I.	67.4	151	269	606	1,080	1,684	2,430	4,300	6,735	9,700	17,216	26,900	38,780
				II.	2.340	1.56	1.17	.780	.585	.468	.390	.292	.234	.195	.146	.117	.098
11	.156	.0323	.0605	I.	68.7	154	274	617	1,100	1,715	2,475	4,380	6,860	9,880	17,540	27,400	39,500
				II.	2.420	1.616	1.210	.808	.605	.484	.404	.303	.242	.202	.151	.121	.101
11.2	.162	.0321	.0623	I.	69.9	156.8	279	628	1,120	1,746	2,520	4,460	6,985	10,060	17,858	27,900	40,220
				II.	2.492	1.664	1.246	.832	.623	.498	.416	.312	.249	.208	.156	.125	.104
11.4	.167	.0320	.0641	I.	71.1	159.5	284	640	1,140	1,778	2,565	4,540	7,110	10,240	18,176	28,400	40,940
				II.	2.564	1.712	1.282	.856	.641	.513	.428	.321	.256	.214	.160	.128	.107
11.6	.173	.0318	.0661	I.	72.3	162.3	289	651	1,160	1,810	2,610	4,620	7,235	10,420	18,494	28,900	41,660
				II.	2.644	1.768	1.322	.884	.661	.529	.442	.331	.264	.221	.165	.132	.111
11.8	.18	.0317	.0684	I.	73.6	165	294	662	1,180	1,841	2,655	4,700	7,360	10,600	18,812	29,400	42,380
				II.	2.736	1.824	1.368	.912	.684	.547	.456	.342	.274	.228	.171	.137	.114
12	.186	.0316	.0703	I.	74.9	168	299	673	1,200	1,872	2,700	4,780	7,485	10,780	19,130	29,900	43,100
				II.	2.812	1.880	1.406	.940	.703	.562	.470	.352	.281	.235	.176	.141	.118
12.2	.192	.0314	.0723	I.	76.1	170.8	304	685	1,220	1,903	2,745	4,860	7,610	10,960	19,454	30,380	43,800
				II.	2.892	1.928	1.446	.964	.723	.578	.482	.362	.289	.241	.181	.145	.121
12.4	.198	.0313	.0742	I.	77.3	173.5	309	696	1,240	1,934	2,790	4,940	7,735	11,140	19,778	30,860	44,500
				II.	2.968	1.984	1.484	.992	.742	.594	.496	.371	.297	.248	.186	.148	.124
12.6	.204	.0312	.0762	I.	78.7	176.1	314	707	1,260	1,965	2,835	5,020	7,860	11,320	20,102	31,340	45,200
				II.	3.048	2.032	1.524	1.016	.762	.610	.508	.381	.305	.254	.191	.152	.127
12.8	.211	.0311	.0785	I.	79.9	179	318	718	1,280	1,996	2,880	5,100	7,985	11,500	20,426	31,820	45,900
				II.	3.140	2.096	1.570	1.048	.785	.628	.524	.393	.314	.262	.196	.157	.131
13	.218	.0310	.0810	I.	81.1	182	323	730	1,300	2,027	2,925	5,180	8,100	11,680	20,750	32,300	46,600
				II.	3.240	2.160	1.620	1.080	.810	.648	.540	.405	.324	.270	.203	.162	.135
13.2	.225	.0309	.0833	I.	82.3	185	328	741	1,320	2,059	2,970	5,260	8,225	11,854	21,070	32,800	47,320
				II.	3.332	2.224	1.666	1.112	.833	.666	.556	.416	.333	.278	.209	.167	.139

TABLE X.—(*Continued*).

Velocity inches per sec.	I.—Value of $\frac{v^2}{2g}$ inches.	II.—Value of ρ.	III.—Value of $\frac{\rho v^2}{64.4}$	Size of pipe.	¼″	⅜″	½″	¾″	1″	1¼″	1½″	2″	2½″	3″	4″	5″	6″
				Area □″	.0491	.110	.196	.442	.785	1.23	1.77	3.14	4.91	7.07	12.57	19.6	28.27
				Value of $\frac{10}{d}$	40	26.65	20	13.33	10	8	6.66	5	4	3.33	2.5	2	1.67
				Value of $\frac{3,600 \times a}{28.33}$	6.23	14	24.9	56.2	99.7	156	225	398	623	898	1,595	2,490	3,590
13.4	.232	.0308	.0858	I.	83.7	187.5	333	752	1,340	2,090	3,015	5,340	8,350	12,028	21,390	33,300	48,040
				II.	3.432	2.288	1.716	1.144	.858	.686	.572	.429	.343	.286	.215	.172	.143
13.6	.239	.0306	.0871	I.	84.9	190.5	338	763	1,360	2,120	3,060	5,420	8,475	12,202	21,710	33,800	48,760
				II.	3.484	2.328	1.742	1.164	.871	.697	.582	.436	.348	.291	.218	.174	.146
13.8	.246	.0305	.0900	I.	86.1	193	343	774	1,380	2,151	3,105	5,500	8,600	12,376	22,030	34,300	49,480
				II.	3.60	2.40	1.80	1.20	.900	.720	.600	.450	.360	.300	.225	.180	.150
14	.254	.0303	.0925	I.	87.3	196	348	785	1,400	2,182	3,150	5,580	8,725	12,550	22,350	34,800	50,200
				II.	3.70	2.464	1.850	1.232	.925	.740	.616	.463	.370	.308	.231	.185	.154
14.2	.261	.0302	.0950	I.	88.6	198.5	353	797	1,420	2,213	3,195	5,660	8,850	12,730	22,660	35,300	50,920
				II.	3.80	2.536	1.900	1.268	.950	.760	.634	.475	.380	.317	.238	.190	.158
14.4	.269	.0301	.0970	I.	89.9	201	358	809	1,440	2,244	3,240	5,790	8,975	12,910	22,970	35,800	51,640
				II.	3.880	2.584	1.940	1.292	.970	.776	.646	.485	.388	.322	.243	.194	.162
14.6	.276	.0300	.0992	I.	91	204	363	820	1,460	2,275	3,285	5,820	9,100	13,090	23,280	36,300	52,360
				II.	3.968	2.648	1.984	1.324	.992	.794	.662	.496	.397	.331	.248	.198	.166
14.8	.284	.0299	.102	I.	92.3	207	368	831	1,480	2,306	3,330	5,900	9,225	13,270	23,590	36,800	53,080
				II.	4.08	2.720	2.04	1.36	1.02	.816	.680	.510	.408	.340	.255	.204	.170

15	.291	.0298	.104	I.	93.5	210	374	842	1,500	2,338	3,375	5,980	9,350	13,450	23,900	37,300	53,800
				II.	4.16	2.796	2.08	1.398	1.04	.832	.694	.520	.416	.347	.260	.208	.174
15.2	.299	.0297	.1065	I.	94.9	213	378	853	1,520	2,370	3,420	6,060	9,475	13,630	24,220	37,800	54,540
				II.	4.260	2.840	2.130	1.420	1.065	.852	.710	.533	.426	.355	.266	.213	.178
15.4	.307	.0295	.1087	I.	96.1	216	383	864	1,540	2,401	3,465	6,140	9,600	13,810	24,540	38,300	55,280
				II.	4.348	2.896	2.174	1.448	1.087	.870	.724	.544	.435	.362	.272	.217	.181
15.6	.315	.0294	.1110	I.	97.3	218	388	875	1,560	2,432	3,510	6,220	9,725	13,990	24,860	38,800	56,020
				II.	4.440	2.96	2.22	1.48	1.110	.888	.740	.555	.444	.370	.278	.222	.185
15.8	.323	.0293	.1135	I.	98.7	221	393	886	1,580	2,463	3,555	6,300	9,850	14,170	25,180	39,300	56,760
				II.	4.540	3.02	2.27	1.512	1.135	.908	.756	.568	.454	.378	.284	.227	.189
16	.331	.0292	.1160	I.	99.9	224	398	898	1,600	2,494	3,600	6,380	9,975	14,350	25,500	39,800	57,500
				II.	4.640	3.096	2.320	1.548	1.160	.928	.774	.580	.464	.387	.290	.232	.193
16.2	.339	.0291	.1182	I.	101	227	403	910	1,620	2,525	3,645	6,460	10,100	14,530	25,820	40,300	58,200
				II.	4.728	3.160	2.364	1.580	1.182	.946	.790	.591	.473	.395	.296	.236	.198
16.4	.348	.0290	.1210	I.	102.4	230	408	921	1,640	2,556	3,690	6,540	10,225	14,710	26,140	40,800	58,900
				II.	4.840	3.224	2.420	1.612	1.210	.968	.806	.605	.484	.403	.303	.242	.202
16.6	.356	.0289	.1232	I.	103.4	232	413	932	1,660	2,587	3,735	6,620	10,350	14,890	26,460	41,300	59,600
				II.	4.928	3.296	2.464	1.648	1.232	.986	.824	.616	.493	.412	.308	.246	.206
16.8	.364	.0288	.1255	I.	104.9	235	418	943	1,680	2,618	3,780	6,700	10,475	15,070	26,780	41,800	60,300
				II.	5.020	3.344	2.510	1.672	1.255	1.004	.836	.628	.502	.418	.314	.251	.210
17	.373	.0287	.1285	I.	106	238	423	954	1,700	2,649	3,825	6,780	10,600	15,250	27,100	42,300	61,000
				II.	5.14	3.424	2.57	1.712	1.285	1.028	.856	.643	.514	.428	.321	.257	.214
17.2	.380	.0286	.1302	I.	107.3	241	428	965	1,720	2,680	3,870	6,860	10,725	15,430	27,420	42,800	61,720
				II.	5.208	3.480	2.604	1.74	1.302	1.042	.870	.651	.521	.435	.326	.260	.218
17:4	.389	.0285	.1330	I.	108.5	244	433	976	1,740	2,711	3,915	6,940	10,850	15,610	27,740	43,300	62,440
				II.	5.320	3.552	2.66	1.776	1.330	1.064	.888	.665	.532	.444	.333	.266	.222

TABLE X.—(*Continued*).

Velocity inches per sec.	I.—Value of $\frac{v^2}{2g}$ inches.	II.—Value of ρ.	III.—Value of $\rho \frac{v^2}{64.4}$	Size of pipe.	1/4″	3/8″	1/2″	3/4″	1″	1 1/4″	1 1/2″	2″	2 1/2″	3″	4″	5″	6″
				Area □″	.0491	.110	.196	.442	.785	1.23	1.77	3.14	4.91	7.07	12.57	19.6	28.27
				Value of $\frac{10}{d}$	40	26.65	20	13.33	10	8	6.66	5	4	3.33	2.5	2	1.67
				Value of $\frac{3{,}600 \times a}{28.33}$	6.23	14	24.9	56.2	99.7	156	225	398	623	898	1,595	2,490	3,590
17.6	.400	.0284	.137	I.	109.9	246	438	987	1,760	2,742	3,960	7,020	10,975	15,790	28,060	43,800	63,160
				II.	5.52	3.68	2.76	1.84	1.38	1.104	.920	.690	.552	.460	.345	.276	.228
17.8	.409	.0283	.141	I.	111	249	443	998	1,780	2,773	4,005	7,100	11,100	15,970	28,380	44,300	63,880
				II.	5.64	3.76	2.82	1.88	1.41	1.128	.940	.705	.564	.470	.352	.282	.235
18	.419	.0283	.1435	I.	112.3	252	448	1,010	1,800	2,804	4,050	7,180	11,225	16,150	28,700	44,800	64,600
				II.	5.74	3.83	2.87	1.916	1.435	1.148	.958	.718	.574	.479	.359	.287	.239
18.5	.442	.0282	.1515	I.	115.5	259	460	1,040	1,850	2,882	4,170	7,380	11,532	16,600	29,500	46,050	66,350
				II.	6.06	4.04	3.03	2.02	1.515	1.212	1.01	.758	.606	.505	.379	.303	.253
19	.466	.0281	.1580	I.	118.5	266	473	1,065	1,900	2,960	4,280	7,580	11,840	17,050	30,300	47,300	68,100
				II.	6.32	4.216	3.16	2.108	1.58	1.264	1.054	.790	.632	.527	.395	.316	.264
19.5	.492	.0280	.166	I.	121.5	273	485	1,095	1,950	3,038	4,390	7,780	12,145	17,500	31,050	48,500	69,900
				II.	6.64	4.424	3.32	2.212	1.66	1.328	1.106	.83	.664	.553	.415	.332	.277
20	.517	.0278	.1735	I.	125	280	497	1,122	2,000	3,116	4,500	7,980	12,450	17,950	31,800	49,700	71,700
				II.	6.94	4.62	3.47	2.31	1.735	1.398	1.156	.868	.694	.578	.434	.347	.2895
20.5	.543	.0275	.181	I.	128	287	510	1,150	2,050	3,194	4,610	8,180	12,775	18,400	32,650	50,450	73,450
				II.	7.24	4.824	3.62	2.412	1.81	1.448	1.206	.905	.724	.603	.452	.362	.302

21	.569	.0274	.1885	I.	131	294	523	1,180	2,100	3,272	4,720	8,380	13,100	18,850	33,500	51,200	75,200
				II.	7.54	5.032	3.77	2.516	1.885	1.508	1.258	.943	.754	.629	.471	.377	.314
21.5	.597	.0272	.1970	I.	134	301	535	1,205	2,150	3,350	4,840	8,580	13,400	19,300	34,250	52,950	77,050
				II.	7.88	5.256	3.94	2.628	1.97	1.576	1.314	.985	.788	.657	.493	.394	.328
22	.625	.0271	.201	I.	137	308	548	1,235	2,200	3,428	4,950	8,780	13,700	19,750	35,000	54,700	78,900
				II.	8.04	5.36	4.02	2.68	2.01	1.608	1.34	1.01	.804	.670	.502	.402	.342
22.5	.653	.0269	.2130	I.	140	315	560	1,265	2,250	3,506	5,070	8,980	14,010	20,200	35,850	55,950	80,700
				II.	8.52	5.68	4.26	2.84	2.13	1.704	1.42	1.065	.852	.710	.533	.426	.356
23	.682	.0268	.222	I.	143.5	322	572	1,290	2,300	3,584	5,170	9,180	14,320	20,650	36,700	57,200	82,500
				II.	8.88	5.92	4.44	2.96	2.22	1.776	1.48	1.11	.888	.740	.555	.444	.370
23.5	.712	.0267	.2305	I.	146.5	329	585	1,320	2,350	3,662	5,280	9,380	14,635	21,100	37,450	58,450	84,250
				II.	9.22	6.14	4.61	3.07	2.31	1.844	1.536	1.15	.922	.768	.576	.461	.385
24	.742	.0266	.2390	I.	150	336	597	1,345	2,400	3,740	5,400	9,580	14,950	21,550	38,200	59,700	86,000
				II.	9.56	6.38	4.78	3.19	2.39	1.912	1.594	1.195	.956	.797	.598	.478	.399
25	.807	.0263	.2570	I.	156	350	622	1,405	2,500	3,900	5,620	9,970	15,580	22,450	39,800	62,200	89,700
				II.	10.28	6.86	5.14	3.43	2.57	2.056	1.714	1.285	1.028	.857	.643	.514	.428
26	.873	.0261	.2755	I.	162	364	647	1,460	2,600	4,050	5,850	10,350	16,200	23,350	41,400	69,700	93,100
				II.	11.02	7.344	5.51	3.67	2.76	2.204	1.84	1.378	1.102	.918	.689	.551	.460
27	.941	.0259	.294	I.	168.5	378	672	1,515	2,700	4,210	6,080	10,750	16,800	24,250	43,000	67,200	96,900
				II.	11.76	7.84	5.88	3.92	2.94	2.352	1.96	1.47	1.176	.98	.735	.588	.490
28	1.012	.0256	.315	I.	175	392	697	1,570	2,800	4,360	6,300	11,150	17,450	25,100	44,600	69,700	100,500
				II.	12.6	8.4	6.3	4.20	3.15	2.52	2.10	1.575	1.26	1.05	.787	.630	.525
29	1.087	.0255	.335	I.	181	406	721	1,630	2,900	4,520	6,520	11,550	18,080	26,000	46,200	72,100	104,000
				II.	13.4	8.96	6.70	4.48	3.35	2.68	2.24	1.675	1.34	1.12	.837	.670	.560
30	1.162	.0253	.355	I.	187	420	747	1,682	3,000	4,675	6,740	11,950	18,700	26,900	47,800	74,600	107,500
				II.	14.2	9.48	7.10	4.74	3.55	2.84	2.37	1.775	1.42	1.185	.887	.710	.593

TABLE X.—(*Continued*).

Velocity inches per sec.	I.—Value of $\frac{v^2}{2g}$ inches.	II.—Value of ρ.	III.—Value of $\frac{\rho v^2}{64.4}$	Size of pipe.	¼″	⅜″	½″	¾″	1″	1¼″	1½″	2″	2½″	3″	4″	5″	6″
				Area □″	.0491	.110	.196	.442	.785	1.23	1.77	3.14	4.91	7.07	12.57	19.6	28.27
				Value of $\frac{10}{d}$	40	26.65	20	13.33	10	8	6.66	5	4	3.33	2.5	2	1.67
				Value of $\frac{3{,}600 \times a}{28.33}$	6.23	14	24.9	56.2	99.7	156	225	398	623	898	1,595	2,490	3,590
31	1.242	.0251	.377	I.	193.5	434	771	1,740	3,100	4,830	6,970	12,350	19,310	27,800	49,400	77,100	111,300
				II.	15.08	10.08	7.54	5.04	3.77	3.016	2.52	1.885	1.508	1.26	.942	.754	.630
32	1.325	.0249	.399	I.	200	448	796	1,795	3,200	4,980	7,200	12,750	19,990	28,700	51,000	79,600	114,900
				II.	15.96	10.64	7.98	5.32	3.99	3.192	2.66	1.99	1.596	1.33	.997	.798	.666
33	1.405	.0248	.421	I.	206	462	821	1,850	3,300	5,140	7,420	13,150	20,600	29,600	52,600	82,100	118,400
				II.	16.84	11.24	8.42	5.62	4.21	3.37	2.81	2.11	1.68	1.41	1.052	.842	.703
34	1.492	.0246	.440	I.	212	476	846	1,910	3,400	5,300	7,650	13,540	21,200	30,500	54,200	84,600	122,000
				II.	17.76	11.84	8.88	5.92	4.44	3.55	2.96	2.22	1.78	1.48	1.11	.888	.740
35	1.582	.0245	.468	I.	218	490	870	1,965	3,500	5,460	7,880	13,940	21,800	31,400	55,700	87,100	125,500
				II.	18.72	12.48	9.36	6.24	4.68	3.74	3.12	2.34	1.87	1.56	1.17	.936	.781
36	1.673	.0243	.493	I.	225	504	895	2,020	3,600	5,610	8,100	14,340	22,400	32,300	57,300	89,600	129,100
				II.	19.72	13.12	9.86	6.56	4.93	3.94	3.28	2.46	1.97	1.64	1.23	.986	.823
38	1.867	.0241	.544	I.	237	532	945	2,130	3,800	5,920	8,540	15,140	23,700	34,100	60,500	94,500	136,400
				II.	21.76	14.48	10.88	7.24	5.44	4.35	3.62	2.72	2.18	1.81	1.36	1.088	.908
40	2.067	.0237	.595	I.	250	560	995	2,240	4,000	6,230	9,000	15,990	24,900	35,800	63,700	99,500	143,500
				II.	23.80	15.88	11.9	7.94	5.95	4.76	3.97	2.98	2.38	1.985	1.487	1.190	.992

42	2.280	.0236	.650	I.	262	587	1,045	2,360	4,200	6,540	9,450	16,730	26,200	37,700	67,000	104,500	150,700
				II.	26.0	17.36	13.0	8.68	6.50	5.20	4.34	3.25	2.60	2.17	1.625	1.30	1.085
44	2.500	.0234	.708	I.	275	615	1,095	2,470	4,400	6,860	9,900	17,530	27,400	39,500	70,000	109,500	158,000
				II.	28.32	18.88	14.16	9.44	7.08	5.66	4.72	3.54	2.83	2.36	1.77	1.416	1.18
46	2.730	.0232	.765	I.	287	642	1,145	2,580	4,600	7,170	10,350	18,330	28,700	41,200	73,200	114,500	165,000
				II.	30.6	20.4	15.3	10.2	7.65	6.12	5.10	3.73	3.06	2.55	1.862	1.53	1.275
48	2.980	.0230	.828	I.	300	672	1,195	2,690	4,800	7,480	10,800	19,130	29,900	43,100	76,500	119,500	172,000
				II.	33.12	21.84	16.56	10.92	8.28	6.62	5.46	4.14	3.31	2.76	2.07	1.66	1.38
54	3.760	.0225	1.025	I.	337	755	1,343	3,030	5,400	8,410	12,150	21,500	33,700	48,400	86,000	134,200	193,800
				II.	41.0	27.36	20.5	13.68	10.25	8.20	6.84	5.13	4.10	3.42	2.56	2.05	1.71
60	4.650	.0221	1.245	I.	374	840	1,493	3,365	6,000	9,350	13,500	23,900	37,400	53,800	95,700	149,200	215,500
				II.	49.8	33.2	24.9	16.6	12.45	9.96	8.30	6.225	4.98	4.15	3.11	2.49	2.08
66	5.630	.0217	1.480	I.	411	922	1,642	3,700	6,600	10,300	14,850	26,300	41,100	59,200	105,000	168,200	236,500
				II.	59.2	39.44	29.6	19.72	14.8	11.84	9.86	7.40	5.92	4.93	3.70	2.96	2.47
72	6.700	.0214	1.740	I.	450	1,006	1,792	4,040	7,200	11,230	16,200	28,700	44,800	64,500	114,800	179,000	258,000
				II.	69.6	46.4	34.8	23.2	17.4	13.92	11.6	8.70	6.96	5.8	4.35	3.48	2.80
84	9.210	.0209	2.310	I.	523	1,175	2,092	4,710	8,370	13,100	18,900	33,400	52,300	75,400	134,000	209,000	301,000
				II.	92.4	61.7	46.2	30.8	23.1	18.48	15.75	11.55	9.24	7.69	5.78	4.62	3.85
96	11.90	.0205	2.920	I.	598	1,343	2,387	5,380	9,570	14,970	21,600	38,200	59,100	86,100	153,000	239,000	344,000
				II.	116.8	77.9	58.4	38.9	29.2	23.36	19.4	14.6	11.68	9.72	7.3	5.84	4.87
108	15.05	.0201	3.620	I.	673	1,510	2,685	6,058	10,770	16,850	24,300	43,000	67,300	97,000	172,200	269,000	387,500
				II.	144.8	96.5	72.4	48.3	36.2	28.96	24.1	18.1	14.48	12.05	9.05	7.24	6.04
120	18.6	.0198	4.420	I.	747	1,680	2,983	6,730	11,960	18,710	27,000	47,800	74,700	107,800	191,300	299,000	430,500
				II.	176.8	118	88.4	59	44.2	35.36	29.4	22.1	17.68	14.75	11.05	8.84	7.38
132	22.5	.0196	5.290	I.	821	1,848	3,281	7,400	13,150	20,590	29,700	52,500	82,200	118,500	210,300	328,500	474,000
				II.	211.6	141	105.8	70.5	52.9	42.32	35.2	26.45	21.16	17.6	13.23	10.58	8.82
144	26.7	.0194	6.210	I.	897	2,016	3,580	8,080	14,350	22,450	32,400	57,300	89,700	129,200	229,700	358,500	517,000
				II.	248.4	166	124.2	83	62.1	49.68	41.3	31.05	24.84	20.7	15.53	12.42	10.37

TABLE XI. (see p. 188, Chapters XIV., XVII.).

HALDANE'S INVESTIGATIONS OF NATURAL INTERCHANGE OF AIR IN BUILDINGS.

Description of Room.	Remarks.	Hours since candle lighted.	Temperature Fahrenheit.		Number of interchanges per hour.
			Out.	In.	
Room A.—Capacity, 1,390 cubic ft. and 9.3 ft. high. Bedroom on first floor. One outside wall of brick. Walls and ceiling papered.	Flap of fire-place open. Breeze scarcely perceptible throughout experiment.	1.1			1.24
		4.1	57.6	60.3	0.91
	Chimney draught = 620 cubic ft. per hour.	5.6			1.0
Same Room	Flap open. Very slight breeze.	2.7	48.2	62	1.0
	Chimney draught = 700 cubic ft. per hour.	3.7			1.11
Same Room	Flap closed. Breeze scarcely perceptible throughout experiment.	3.1	62	63	0.37
		4.1			.295
		5.1			.345
		6.1			0.475
			55.9	64	
		9.6			.155
		11.6	51	64	.40

Same Room	Flap closed. Strong wind throughout experiment.	2.0	58	74.3	0.55
		3.0			.665
		4.0	56	64.4	0.665
		6.0			0.715
		7.5			0.77
		8.5			0.77
	Flap opened	11.0	51.8	64.9	1.43
	Chimney draught = 2,700 cubic ft. per hour.	12.5			1.66
Same Room	Flap closed. Moderate breeze	2.0	60.5	64.9	1.11
		3.0			0.77
	Breeze now very slight ...	5.5			0.455
		6.5			.587
	No breeze	11.5	51.2	65.3	.416
	Lower sash of window now raised 7 inches.	13			1.43
		14	50.4	63.3	1.25

HALDANE'S INVESTIGATIONS OF NATURAL INTERCHANGE OF AIR IN BUILDINGS—(*Continued*).

Description of Room.	Remarks.	Hours since candle lighted.	Temperature, Fahrenheit. Out.	Temperature, Fahrenheit. In.	Number of interchanges per hour.
Same Room	Fire burning brightly in grate. Slight breeze.	1.3	56.3	66.9	5.0
	Chimney draught = 4,450 cubic ft. per hour.	2.0	56.3	66.9	2.5
	Fire much lower. Draught = 3,260 cubic ft. per hour.	3.6	56.8	65.8	2.5
Room B.—786 cubic ft. Attic of irregular shape, on second floor. One window. Walls papered.	Wind imperceptible throughout experiment.	1.5	51.8	64	0.555
		2.3			0.555
		3.2	50.8	64	0.835
Room C.—1,100 cubic ft., 11.5 ft. high and nearly square. A laboratory room on ground floor. One window and one outside wall of sandstone, one inside wall of sandstone, and other two of wood and plaster. Walls not papered. One fixed ventilator at roof, opening about 24 square in. Left open through both experiments.	Wind scarcely perceptible ...	1.0	52.7	56.3	.345
		2.0			0.765
		3.25			0.765
		6.0			0.665
		8.0	48.2	56.5	0.665

Same Room	Strong wind throughout experiment.	1.75	58	63	4.0
		3.3			3.33
		4.0			4.0
Room D.—5,600 cubic ft., 11.5 ft. high and nearly square. Laboratory room on ground floor. Two double windows, and one outside wall of sandstone and one of wood and plaster. No fire-place. Room not papered.	Wind scarcely perceptible. Simultaneous with first experiments on *Rooms C* and *E*.	0.6			
		1.25	53.3	56.8	.238
		2.7			.217
		4.0			.192
		4.7			.208
		7.5			.222
		9.25	48.6	57.2	.243
Same Room	Strong wind. Simultaneous with second experiments on *Rooms C* and *E*.	1.6			.527
		2.4			.625
		3.2			.555
		3.8	58	64	.713

HALDANE'S INVESTIGATIONS OF NATURAL INTERCHANGE OF AIR IN BUILDINGS—(*Continued*).

Description of Room.	Remarks.	Hours since candle lighted.	Temperature, Fahrenheit.		Number of interchanges per hour.
			Out.	In.	
Room E.—18,800 cubic ft., 11.5 ft. high and 70 × 24 ft. A long laboratory room on ground floor. Windows and four doors. 70% of wall is to outside and of sandstone. Otherwise like *Room D.*	Wind scarcely perceptible. Simultaneous with first experiments on *Rooms D* and *C.* At roof.	0.5	53.3	56.3	
		2.2			0.5
		3.5			0.4
		5.2			0.476
		7.2	48.6	56.3	0.555
Same Room	Strong wind. Simultaneous with second experiments on *Rooms C* and *D.*	1.4			0.5
		2.0			0.666
		2.8			0.666
	Wind now less strong ...	3.6	58	63.5	0.555
	Wind stronger again ...	4.2			0.625

Room F.—13,300 cubic ft., 15 ft. high and nearly square. A large ground floor room in Auchterarder, used as a gymnasium. One large open fire-place and two doors. Walls of stone.	Gentle breeze throughout experiment. Very perceptible draught up chimney, but fire not lit. Simultaneous with first experiment on *Room G.*	0.0	60	62.2	
		3.0			.345
		4.8			0.527
		6.2			0.527
		6.7			0.527
Room G.—75,000 cubic ft., 79 × 40 ft. and 30 ft. high in centre. Town Hall of Auchterarder. Gallery behind, about 15 ft. high in centre. Stone walls. Sloping roof with sky-lights. Ventilators closed.	Gentle easterly breeze through experiment.	0.0	57.8	59.8	
		3.8	57.2	62.2	.435
		4.0			.417
		5.7	55.9	62.6	.476
		7.5			.455
		8.2			.500
		8.5	54	62.6	.476
Same Room	Wind scarcely perceptible ...	0.0	62	60.8	
		2.0			.303
		2.8	61.7	63.5	.400

HALDANE'S INVESTIGATIONS OF NATURAL INTERCHANGE OF AIR IN BUILDINGS—(*Continued*).

Description of Room.	Remarks.	Hours since candle lighted.	Temperature, Fahrenheit.		Number of interchanges per hour.
			Out.	In.	
Same Room	Middle of gallery	3.0	61.7	64.8	
		3.7			.385
	Platform in front of hall ...	4.1			.370
		4.9	63.2	65.3	.333
	Gallery	5.2		64	
		5.6			.313
		6.6			.286
Room H.—72,000 cubic ft., 57½ × 46½ ft. and 28 ft. high in centre. Free Church, Auchterarder. Gallery round three sides, and about 15 ft. high in centre. Lighted by windows. Ventilated by openings measuring about 46 square ft. in all, and communicating with loft below slates. Side windows.	Gentle N.W. breeze. Air entering by loosely fitting windows on west side.	0.0	54.8	53	
		1.2			.357
		2.1			.323
		3.1			.303
		4.5			.376
		5.5			.455

TABLE XI.*a* (see p. 169).

P. J. KIRKBY'S TABLE SHOWING PROBABLE MAXIMUM VITIATION OF AIR IN A ROOM.

E = actual excess of impurity (inner — outer air) = $(K - a)$.
E_0 = excess with no ventilation $\frac{nbt \times 10{,}000}{c}$
T = time for completely changing air.
t = time room occupied.

The maximum to which E will rise is found by multiplying E_0 by the value of $\frac{T}{t}$ corresponding to the observed and calculated value of $\frac{E}{E_0}$

Value of $\frac{E}{E_0}$	Value of $\frac{T}{t}$	Value of $\frac{E}{E_0}$	Value of $\frac{T}{t}$	Value of $\frac{E}{E_0}$	Value of $\frac{T}{t}$
.95	10.0	.72	1.43	.50	.62
.93	7.5	.69	1.25	.48	.59
.90	5.0	.66	1.1	.45	.53
.87	3.3	.63	1.0	.43	.50
.85	3.0	.61	.91	.40	.45
.82	2.5	.58	.83	.37	.40
.79	2.0	.56	.77	.35	.37
.77	1.8	.54	.71	.30	.31
.75	1.67	.52	.67	.25	.25

TABLE XII. (see page 87).

RIETSCHEL'S VALUES OF ζ FOR AIR IN CHANNELS.

The co-efficients for single resistance where actual resistance $= \zeta \frac{v^2}{2\,g}$ in feet of fluid column.

	Value of ζ
Change of direction.	
A right-angled sharp elbow	1.5
A right-angled round elbow	1.0
A 135″ bend	0.6
Easy bend with radius at least 5 times flue diameter	0
Actual obstructions.	
An open valve where the frame is flush with the wall of the flue	0
Gratings.	
For a grating whose free area is equal to the section of the flue.	
When the free area is half the total area, *i.e.*, total area = 2 × flue area	1.5
When the free area is 0.2 of the total area of the grating	2.0
For a grating whose free area is 1.5 times the section of the flue.	
When the free area = 0.5 × total area	0.75
When the free area = 0.2 × total area	1
For a wide-meshed wire grating	0
For a wire grating whose free area = the flue section and when $\frac{\text{free area}}{\text{total area}}$ is not under 0.6	0.6
Do. do., but whose free area = 1.5 flue section free area of grating = 0.6 or more × total area	0.3
Change of section.	
Small smooth changes of section may be neglected.	
For large sudden changes of section from f' sq. ft. to f sq. ft., *e.g.*, heating chambers or dust chambers where the velocity does not become practically = 0, calculated with reference to the larger velocity	$\zeta = (\frac{f}{f_1} - 1)$

TABLE XII.—*(Continued)*.

	Increase.	Value of ζ
Or calculated with reference to the smaller velocity		$\zeta = (1 - \frac{f_1}{f})^2$
(*a*) Where branch is taken from side of channel, round or square		1.1
(*b*) Rectangular section—		
(*a*) Long side diverted		1.1
(*b*) Short side diverted		$\zeta = 1.1 + \frac{b-a}{2})$
Two right-angle knees *a*, distance *e* apart from one another.		
The foregoing values have to be increased *for each knee* as follows :—		
Where $e \geqq 3a$	80%	
$e > 3a < 5a$	50%	
$e > 5a < 8a$	30%	
$e > 8a$	0%	
A sharp turning off at an angle of 135″		0.3
A right-angled bend with round or rectangular section. r = radius of centre line, s = width in direction of radius.		
Where $r = s$		0.25
$r > s$		0.20
$r > 2s$		0.15
$r > 4S$		0.12
$r > 5S$		0.07
$r > 6S$		0.0
A 135″ bend $r \leqq 2S$		0.15
A retroflex bend		
$r \geqq 3S$		0.4
$R > 3S < 8S$		0.25
$r > 8s < 12S$		0.1
$r > 12s$		0.0

The above values apply equally to built or metal channels.

TABLE XII.*a*.

Weisbach's (I.) Values of ρ calculated from the formula

$$\rho = .01439 + \frac{.0595}{\sqrt{v}} \quad (v \text{ in inches per sec.})$$

(II.) Values of $\frac{v^2}{2\ g}$

(III.) Values of $\rho\frac{v^2}{64.4}$

The values in Column III. multiplied by $\frac{\text{length of pipe (feet)}}{\text{diameter of pipe (inches)}}$ give resistance in inches of water column.

Velocity in inches per sec.	Value of ρ	Value of $\frac{v^2}{2\ g}$(inches).	Value of $\rho\frac{v^2}{64.4}$	Velocity in inches per sec.	Value of ρ	Value of $\frac{v^2}{2\ g}$ (inches).	Value of $\rho\frac{v^2}{64.4}$
1	.0739	.00129	.001141	6.8	.0372	.0597	.0266
1.2	.0686	.0020	.001645	7	.0369	.0634	.0281
1.4	.0645	.0029	.00241	7.2	.0366	.0670	.0294
1.6	.0613	.0033	.002428	7.4	.0363	.071	.0309
1.8	.0587	.0041	.002882	7.6	.0360	.074	.03195
2	.0564	.00516	.003480	7.8	.0357	.078	.0333
2.2	.0545	.0063	.00412	8	.0354	.0827	.0351
2.4	.0528	.0077	.00488	8.2	.0351	.087	.0366
2.6	.0513	.0082	.00505	8.4	.0349	.091	.0381
2.8	.0499	.01	.005985	8.6	.0346	.095	.0394
3	.0487	.0116	.00677	8.8	.0344	.10	.0413
3.2	.0477	.013	.00742	9	.0342	.1045	.0428
3.4	.0467	.015	.00840	9.2	.0340	.109	.0445
3.6	.0457	.0165	.00904	9.4	.0337	.114	.0460
3.8	.0449	.0185	.00996	9.6	.0336	.118	.0475
4	.0441	.0207	.01093	9.8	.0334	.124	.0497
4.2	.0434	.0227	.01182	10	.0332	.129	.0513
4.4	.0427	.0245	.01255	10.2	.0330	.134	.0530
4.6	.0421	.0273	.01375	10.4	.0329	.139	.0548
4.8	.0415	.0292	.01453	10.6	.0327	.145	.0568
5	.0409	.0323	.01585	10.8	.0325	.150	.0585
5.2	.0404	.0350	.01697	11	.0323	.156	.0605
5.4	.0400	.0373	.01790	11.2	.0321	.162	.0623
5.6	.0395	.040	.01895	11.4	.0320	.167	.0641
5.8	.0391	.0432	.02025	11.6	.0318	.173	.0661
6	.0387	.0465	.02160	11.8	.0317	.18	.0684
6.2	.0383	.0492	.02262	12	.0316	.186	.0703
6.4	.0379	.0525	.02390	12.2	.0314	.192	.0723
6.6	.0376	.0593	.0245	12.4	.0313	.198	.0742

TABLE XII.*a*—(*Continued*).

Velocity in inches per sec.	Value of ρ	Value of $\frac{v^2}{2\ g}$ (inches).	Value of $\rho\ \frac{v^2}{64.4}$	Velocity in inches per sec.	Value of ρ	Value of $\frac{v^2}{2\ g}$ (inches).	Value of $\rho\ \frac{v^2}{64.4}$
12.6	.0312	.204	.0762	20	.0277	.517	.1735
12.8	.0311	.211	.0785	20.5	.0275	.543	.181
13	.0310	.218	.0810	21	.0274	.569	.1885
13.2	.0309	.225	.0833	21.5	.0272	.597	.1970
13.4	.0308	.232	.0858	22	.0271	.625	.201
13.6	.0306	.238	.0871	22.5	.0269	.653	.2130
13.8	.0305	.246	.0900	23	.0268	.682	.2220
14	.0303	.254	.0925	23.5	.0267	.712	.2305
14.2	.0302	.261	.0950	24	.0266	.742	.2390
14.4	.0301	.269	.0970	25	.0263	.807	.2570
14.6	.0300	.276	.0992	26	.0261	.873	.2755
14.8	.0299	.284	.102	27	.0259	.941	.294
15	.0298	.291	.104	28	.0256	1.012	.315
15.2	.0297	.299	.1065	29	.0255	1.087	.335
15.4	.0295	.307	.1087	30	.0253	1.162	.355
15.6	.0294	.315	.1110	31	.0251	1.242	.377
15.8	.0293	.323	.1135	32	.0249	1.325	.399
16	.0292	.331	.1160	33	.0248	1.405	.421
16.2	.0291	.339	.1182	34	.0246	1.492	.444
16.4	.0290	.348	.1210	35	.0245	1.582	.468
16.6	.0289	.356	.1232	36	.0243	1.673	.493
16.8	.0288	.364	.1255	38	.0241	1.867	.544
17	.0287	.373	.1285	40	.0237	2.067	.595
17.2	.0286	.380	.1302	42	.0236	2.280	.650
17.4	.0285	.389	.1330	44	.0234	2.500	.708
17.6	.0284	.400	.138	46	.0232	2.730	.765
17.8	.0283	.409	.141	48	.0230	2.980	.828
18	.0283	.419	.1435	54	.0225	3.760	1.025
18.5	.0280	.442	.1515	60	.0221	4.650	1.245
19	.0280	.466	.1580	66	.0217	5.630	1.480
19.5	.0281	.492	.166	72	.0214	6.700	1.740

TABLE XIII. (see page 505).

LENGTH OF STRAIGHT PIPE WHOSE RESISTANCE = $\frac{v^2}{2\,g}$ AT DIFFERENT VELOCITIES.

Velocity $\frac{\text{ins.}}{\text{secs.}}$	2	3	6	9	12	18	24	36	48
Value of $\frac{v^2}{2\,g}$ ins.	.00518	.0116	.0466	.105	.186	.419	.745	1.68	2.99
Size of pipe.									
½″	.745	.865	1.08	1.23	1.325	1.46	1.56	1.71	1.81
¾″	1.12	1.30	1.62	1.84	1.98	2.18	2.34	2.56	2.74
1″	1.49	1.725	2.16	2.46	2.64	2.91	3.12	3.41	3.61
1¼″	1.865	2.15	2.70	3.07	3.32	3.64	3.90	4.27	4.53
1½″	2.24	2.59	3.24	3.67	3.96	4.38	4.69	5.13	5.49
2″	3.52	3.44	4.32	4.90	5.29	5.84	6.21	6.83	7.23
2½″	3.73	4.31	5.41	6.15	6.62	7.30	7.80	8.54	9.05
3″	4.47	5.18	6.49	7.35	7.92	8.75	9.35	10.25	10.95
4″	5.96	6.90	8.65	9.81	10.57	11.7	12.5	13.7	14.4
5″	7.45	8.65	10.8	12.3	13.2	14.6	15.6	17.1	18.0
6″	8.92	10.3	12.9	14.7	15.8	17.5	18.7	20.4	21.7

TABLE XIV. (see Chapter XIV.).

VALUES OF *K* FOR WINDOWS, WALLS, ROOFS, FLOORS, Etc. (RIETSCHEL).

Nature of Structure.	Thickness, inches.	Value of *K* British units.	Equivalent of 1 sq. ft. in sq. ft. of glass.
WINDOWS.			
Single window, ordinary glass ...	—	1.08	1.0
Single window, plate glass	—	1.02	.945
Double window	—	.45	.417
SKYLIGHTS.			
Single, with outer air over	—	1.04	.964
Single, with roof space or garret over	—	.74	.685
Double, with outer air over	—	.48	.444
Double, with garret over	—	.43	.398
WALLS.			
Brickwork	4½	.49	.454
	9	.37	.343
	14	.281	.260
	18	.235	.218
	21	.21	.195
	24	.193	.179
	30	.164	.152
	36	.140	.130
	40	.125	.116
Brickwork with 1″ plaster layer ...	4½	.44	.408
	9	.33	.306
	14	.257	.238
	18	.222	.206
	21	.201	.186
	24	.185	.171
	30	.158	.146
	36	.135	.125
	40	.120	.111
Brickwork with air space	9	.295	.273
	14	.232	.215
	18	.198	.183
	21	.180	.167
	24	.167	.155
	30	.144	.133
	36	.125	.116
	40	.113	.105
Brickwork, lined internally with wood ⅝″ thick	4½	.36	.333
	9	.28	.260
	14	.215	.199
Ditto with ⅜″ wood	4½	.41	.380
	9	.31	.287
	14	.225	.208

Table XIV.—*(Continued).*

Nature of Structure.	Thickness, inches.	Value of K British units.	Equivalent of 1 sp. ft. in sq. ft. of glass.
Brickwork, lined ⅜″ wood externally and internally	4½	.246	.228
	9	.205	.190
	14	.174	.161
Brickwork, lined with ½″ wood externally and internally	4½	.225	.208
	9	.184	.170
	14	.164	.152
Ditto with ⅝″ wood	4½	.205	.190
	9	.164	.152
	14	.143	.133
Sandstone wall	12	.427	.396
	16	.367	.340
	20	.321	.298
	24	.286	.265
	30	.246	.228
	36	.215	.199
	42	.189	.175
	48	.167	.155
Limestone wall	12	.51	.472
	16	.45	.417
	20	.407	.377
	24	.370	.342
	30	.322	.298
	36	.284	.263
	42	.259	.240
	48	.242	.224
Plaster boards	1¼	.76	.705
	1⅝	.698	.647
	2	.656	.610
	2½	.614	.569
Concrete wall	2	.7	.648
	4	.55	.510
	6	.47	.435
	8	.41	.380
	10	.345	.320
	12	.305	.283
Solid wood wall lined with planks ...	4¾	.117	.108
	5	.105	.975
	7	.082	.76
INNER WALLS.			
Brickwork	4½	.45	.417
	9	.307	.284
	14	.246	.228
	20	.205	.190
	25	.164	.152
	30	.143	.133
Wood with plaster layer on both sides	¾	.266	.246
	1	.245	.227
	1¼	.235	.218
	1½	.205	.190

TABLE XIV.—*(Continued)*.

Nature of Structure.	Thickness, inches.	Value of K British units.	Equivalent of 1 sq. ft. in sq. ft. of glass.
Plaster boards	1½	.617	.572
	2	.594	.550
	2½	.572	.530
	3	.518	.480
	3½	.495	.459
	4	.477	.442
FLOORS AND CEILINGS.			
Tiled roof with rafters boarded on one side	—	.35	.324
Lath and plaster ceiling	—	.40	.37
Flat roof of wood joists, covered, zinc or lead on boards with lath and plaster ceiling under	—	.17	.158
Tiled roof with rafters boarded on both sides	—	.18	.167
Reinforced concrete roof with asphalte	5	.30	.278
	6	.28	.260
	7	.27	.250
	9	.25	.232
	11	.23	.213
	13	.21	.194
Joist floor with pugging, lath and plaster ceiling under, cold air above	—	.11	.102
Ditto, with cold air below	—	.06	.0555
Floor, concrete and steel with wood blocks and plaster ceiling, concrete	5	.21	.195
	7	.19	.176
	9	.18	.167
	11	.17	.157
	13	.16	.148
Ordinary joist floor with floor boards and lath and plaster ceiling under, cold air above	—	.17	.157
Ditto, cold air below	—	.07	.065
Joist floor with air space and 6″ of concrete on earth	—	.05	.0463
Solid floor, concrete with ¾″ cement and 4″ wood blocks over concrete	6	.25	.232
	12	.20	.185
Tile roof without lining, otherwise tight	—	.992	.920
Tile roof with lath and plaster ...	—	.328	.304
Corrugated plate without lining ...	—	2.13	1.97
Concrete roof with roof pasteboard, without plaster	3	.535	.495
Ditto with pasteboard and 1″ cork lining, air space and plaster inside	—	.266	.246

TABLE XV.

EMISSION OF HEAT FROM PIPES PER FOOT RUN (RIETSCHEL, ADJUSTED Figs. 93 and 93*a*).

Size of Pipe.	Air Temp.	Mid-temperature $\left(\frac{t_f + t_r}{2}\right)$ of pipes.																	
—	0	60	65	70	75	80	85	90	95	100	105	110	115	120	125	130	135	140	145
—	50	110	115	120	125	130	135	140	145	150	155	160	165	170	175	180	185	190	195
—	55	115	120	125	130	135	140	145	150	155	160	165	170	175	180	185	190	195	200
—	60	120	125	130	135	140	145	150	155	160	165	170	175	180	185	190	195	200	205
—	65	125	130	135	140	145	150	155	160	165	170	175	180	185	190	195	200	205	210
—	70	130	135	140	145	150	155	160	165	170	175	180	185	190	195	200	205	210	215
3/8″	—	23	26	29	31	34	36	39	42	45	47	50	53	56	58	61	64	67	70
1/2″	—	30	33	36	40	43	47	50	54	57	61	64	68	71	75	78	82	86	89
3/4″	—	35	39	43	47	50	54	58	62	66	70	74	78	82	86	90	95	99	103
1″	—	42	46	50	55	60	65	70	74	79	84	89	94	99	104	109	114	119	124
1 1/4″	—	51	57	62	68	74	80	86	91	97	104	110	116	122	128	134	140	147	153
1 1/2″	—	58	64	70	76	82	88	94	101	108	114	121	128	135	142	149	156	163	170
2″	—	68	75	82	89	96	104	110	118	126	134	142	150	159	167	176	184	192	200
2 1/2″	—	85	94	103	111	120	130	139	148	158	168	178	189	199	209	219	229	240	250
3″	—	94	104	113	123	132	142	152	163	173	185	196	207	218	229	240	251	262	274
4″	—	121	133	145	157	170	182	194	208	220	234	247	261	275	288	301	315	328	341
5″	—	146	160	174	188	201	216	230	246	261	277	292	308	323	339	354	370	385	400
6″	—	170	186	200	217	233	250	267	285	301	319	336	354	370	388	406	423	440	458

TABLE XVI.

VALUES OF *K* FOR WATER HEATING SURFACE (MAINLY FROM RIETSCHEL).

(adjusted as Figs. 93 and 98).

(*a*) WATER RADIATION.

	Difference of temperature (surface — air).										
Kind of Surface.	60	70	80	90	100	110	120	130	140	150	160
PIPES.											
Single horizontal pipe—											
½"	2.156	2.255	2.345	2.427	2.500	2.556	2.605	2.645	2.675	2.69	2.70
¾"	2.06	2.145	2.22	2.295	2.36	2.43	2.48	2.52	2.55	2.57	2.58
1"	2.00	2.08	2.15	2.22	2.29	2.35	2.40	2.45	2.48	2.50	2.51
1¼"	1.95	2.02	2.08	2.14	2.2	2.26	2.31	2.35	2.38	2.41	2.43
1½"	1.88	1.94	2.00	2.06	2.12	2.17	2.22	2.27	2.30	2.33	2.35
2"	1.78	1.84	1.90	1.96	2.02	2.07	2.12	2.17	2.21	2.24	2.26
2½"	1.75	1.80	1.86	1.91	1.96	2.01	2.06	2.10	2.14	2.17	2.19
3"	1.72	1.77	1.82	1.87	1.91	1.96	2.00	2.04	2.07	2.10	2.11
4"	1.70	1.74	1.79	1.83	1.87	1.91	1.94	1.97	1.99	2.01	2.02
5"	1.68	1.72	1.75	1.79	1.82	1.85	1.87	1.89	1.91	1.93	1.94
6"	1.66	1.69	1.72	1.75	1.77	1.80	1.82	1.84	1.86	1.87	1.87
Several plain pipes one over the other, distance apart at least = outside diameter of pipe. Total coil not more than 3ft. high—											
Interior diameter up to 1" ...	1.85	1.93	2.01	2.08	2.15	2.19	2.22	2.25	2.29	2.33	2.36
,, above 1" ...	1.44	1.52	1.60	1.67	1.74	1.76	1.78	1.80	1.82	1.84	1.85
As above, but more than 3ft. high—											
Interior diameter up to 1" ...	1.64	1.70	1.75	1.80	1.85	1.87	1.89	1.91	1.93	1.94	1.95
,, above 1" ...	1.32	1.38	1.44	1.49	1.54	1.56	1.58	1.60	1.62	1.63	1.64

TABLE XVI.—*(Continued)*.

Kind of Surface.	Difference of temperature (surface — air).										
	60	70	80	90	100	110	120	130	140	150	160
Battery of ribbed pipes, circular ribs, distance apart ⅝″—											
1 pipe high	0.66	0.74	0.81	0.86	0.91	0.95	0.98	1.01	1.03	1.04	1.05
3 ,, ...	0.53	0.61	0.67	0.73	0.78	0.82	0.85	0.88	0.89	0.91	0.92
6 ,, ...	0.38	0.45	0.52	0.58	0.63	0.67	0.71	0.74	0.77	0.78	0.79
RADIATORS, ETC.											
Flat radiators—											
Up to 3ft. high	1.54	1.59	1.64	1.69	1.74	1.79	1.84	1.88	1.92	1.96	2.00
Above 3ft. high	1.32	1.39	1.44	1.49	1.54	1.59	1.64	1.68	1.72	1.76	1.80
Single section 2-column radiator ...	1.53	1.59	1.64	1.69	1.73	1.76	1.78	1.81	1.83	1.84	1.85
2 ,, ,, ...	1.32	1.38	1.42	1.46	1.50	1.52	1.55	1.56	1.58	1.59	1.60
3 ,, ,, ...	1.26	1.31	1.36	1.40	1.44	1.46	1.49	1.50	1.52	1.53	1.54
4-6 ,, ,, ...	1.20	1.25	1.30	1.34	1.38	1.41	1.43	1.46	1.47	1.48	1.49
6 ,, and over ...	1.13	1.19	1.24	1.28	1.32	1.35	1.38	1.40	1.42	1.43	1.44
Ribbed radiator up to 2ft. high, with vertical ribs not less than 1¾″ centres—											
Height of ribs 0″	1.59	1.65	1.69	1.74	1.77	1.79	1.81	1.82	1.83	1.84	1.85
¾″	1.1	1.16	1.21	1.26	1.30	1.33	1.35	1.36	1.37	1.38	1.38
1½″	0.96	1.02	1.08	1.12	1.16	1.20	1.23	1.25	1.26	1.27	1.28
2″	0.87	0.94	1.00	1.05	1.10	1.14	1.16	1.18	1.19	1.20	1.21
2⅜″	0.77	0.84	0.90	0.95	1.00	1.03	1.06	1.08	1.10	1.11	1.12

TABLE XVII.

HEAT EMITTED BY RADIATORS OF VARIOUS SIZES (RIETSCHEL'S CO-EFFICIENTS ADJUSTED AS FIG. 98).

Surface of Radiator.	Air Temp.	Mid-temperature $\left(\frac{t_f + t_r}{2}\right)$ of radiator.																	
	0	60	65	70	75	80	85	90	95	100	105	110	115	120	125	130	135	140	145
	50	110	115	120	125	130	135	140	145	150	155	160	165	170	175	180	185	190	195
	55	115	120	125	130	135	140	145	150	155	160	165	170	175	180	185	190	195	200
	60	120	125	130	135	140	145	150	155	160	165	170	175	180	185	190	195	200	205
	65	125	130	135	140	145	150	155	160	165	170	175	180	185	190	195	200	205	210
	70	130	135	140	145	150	155	160	165	170	175	180	185	190	195	200	205	210	215
Per 1 (6 sec.)	Approximate B.T.U. emitted.	68	75.5	83.5	91.5	99.5	108	116	124.5	133	141	149.5	158	166	174	182	190.5	199	208
5 (2 sec.)		400	442	485	527	570	613	655	700	745	790	835	880	925	970	1015	1062	1115	1168
10 (3 sec.)		760	845	925	1010	1090	1175	1260	1345	1430	1510	1600	1685	1775	1860	1950	2040	2135	2230
15 (5 sec.)		1080	1207	1327	1454	1575	1700	1830	1957	2080	2205	2325	2460	1585	2705	2825	2950	3020	3210
20 (6 sec.)		1360	1510	1670	1830	1990	2160	2320	2490	2660	2820	2990	3160	3320	3480	3640	3810	3980	4160
25 ,,		1700	1890	2090	2280	2480	2700	2900	3110	3320	3520	3730	3950	4150	4350	4550	4750	4970	5200
30 ,,		2040	2260	2450	2740	2980	3240	3480	3740	3990	4230	4480	4740	4980	5220	5450	5710	5970	6240
35		2380	2640	2920	3200	3480	3780	4060	4350	4650	4930	5230	5530	5810	6080	6370	6670	6960	7270
40		2720	3020	3340	3660	3980	4320	4640	4980	5320	5640	5970	6320	6640	6960	7280	7620	7960	8310
45		3060	3390	3750	4110	4470	4850	4210	5600	5980	6340	6720	7100	7460	7820	8180	8560	8950	9350
50		3400	3770	4170	4570	4970	5400	5800	6220	6650	7050	7470	7900	8300	8700	9100	9450	9950	10400
55		3740	4150	4590	5030	5470	5940	6380	6850	7310	7750	8220	8690	9130	9570	10000	10470	10950	11430
60		4080	4520	5000	5480	5960	6480	6960	7460	7970	8450	8960	9470	9950	10430	10910	11420	11940	12480
65		4420	4900	5420	5940	6460	7010	7530	8090	8640	9150	9700	10270	10790	11300	11820	12380	12920	13510
70		4750	5280	5840	6400	6960	7550	8110	8700	9300	9860	10460	11060	11610	12180	12730	13320	13920	14550
75		5100	5660	6250	6850	7450	8100	8700	9340	9970	10570	11200	11850	12440	13050	13650	14300	14910	15600
80		5440	6030	6680	7310	7950	8630	9280	9960	10630	11290	11960	12630	13280	13910	14550	15230	15910	16620
85		5780	6410	7100	7670	8450	9180	9860	10580	11300	12000	12700	13410	14100	14800	15460	16200	16900	17650
90		6120	6790	7510	8230	8950	9720	10430	11200	11970	12700	13450	14200	14920	15650	16370	17140	17900	18700
95		6460	7170	7930	8680	9450	10260	11000	11820	12620	13400	14200	15000	15760	16510	17300	18100	18900	19750
100		6800	7550	8350	9150	9950	10800	11600	12450	13300	14100	14950	15800	16600	17400	18200	19050	19900	20800
105		7140	7920	8760	9600	10440	11330	12180	13060	13960	14800	15700	16590	17410	18260	19100	20000	20900	21800
110		7480	8300	9180	10060	10950	11890	12760	13700	14610	15500	16440	17380	18250	19130	20000	20950	21900	22860
115		7820	8680	9600	10510	11430	12410	13340	14310	15300	16210	17200	18170	19100	20000	20920	21850	22870	23900
120		8150	9050	10000	10960	11920	12950	13900	14920	15940	16900	17920	18940	19900	20880	21820	22810	23840	24930
125		8500	9430	10420	11420	12420	13500	14500	15550	16600	17600	18650	19720	20730	21720	22730	23800	24830	26000
130		8840	9800	10840	11880	12920	14020	15070	16180	17280	18300	19400	20500	21600	22600	23620	24700	25820	27000

TABLE XVIII.

EFFICIENCY OF COATING COMPOSITIONS (RIETSCHEL).

(See Fig. 97).

Thickness of coating.	Percentage saving on emission from bare pipe.			
	¾″	1″	1½″	2″
Soft brown felt	83.5	86	88.5	89.5
Silk waste over layer of Kieselguhr ...	77.5	80	82	83
Silk waste with air space enclosed in metal casing	75	78	80	81
Kieselguhr with all organic matter burnt out	73	77.5	82	83
Kieselguhr and powdered cork	68.5	72	76	78
Cork matting	63	71	82	88
Asbestos millboard with Kieselguhr filling	57	60.5	64	66
Asbestos string wrapped with asbestos fibre	43.5	46.5	50.5	52
Straw rope and clay	35	40	45	47

TABLE XIX.

VALUES OF *K* FOR VARIOUS LOW PRESSURE STEAM HEATED SURFACES.

Diameter of pipe	½″	¾″	1″	1¼″	1½″	2″	2½″	3″	4″	5″	6″
Value of *K*, horizontal pipe	2.7	2.68	2.65	2.62	2.59	2.56	2.51	2.47	2.42	2.37	2.34
,, vertical pipe	2.8	2.78	2.75	2.72	2.69	2.66	2.61	2.57	2.52	2.47	2.44
,, high pressure pipe ...	2.9	2.88	2.85	2.82	2.79	2.76	2.71	2.67	2.62	2.57	2.54
Heat emitted per foot run by horizontal pipe, low pressures —											
212° to 50°	100	120	147	187	213	258	327	363	462	552	644
212° to 60°	94	113	138	175	200	242	307	340	433	517	604
212° to 70°	87.5	105	128	164	187	226	286	318	405	483	565

Value of *K* for several rows of steam pipe below 1″, less than 3ft. high	2.56
,, ,, ,, above 1″, ,,	2.26
,, ,, ,, below 1″, above 3ft. high ...	2.26
,, ,, ,, above 1″, ,, ...	1.95
Flat steam surface up to 3ft. high	2.45
,, above 3ft. high	2.25

Radiators.	*K*	B.T.U. per sq. ft. 212° to 60°
1 section, 2 col.	2.35	358
2 sections, 2 col.	1.94	295
3 sections, 2 col.	1.84	280
4-6 sections, 2 col.	1.73	265
Over 6 sections, 2 col.	1.64	250

TABLE XX.

VALUE OF *K* FOR HIGH PRESSURE HOT WATER PIPE.

Inside diameter ⅞″. Outside diameter 1.318″.

Diff. of temp. water — air.	60°	70°	80°	90°	100°	110°	120°	130°	140°	150°	160°	180°	200°	220°	240°	260°	280°
Value of *K* ...	1.88	1.99	2.07	2.14	2.20	2.25	2.28	2.30	2.32	2.34	2.35	2.36	2.37	2.38	2.39	2.40	2.41

TABLE XXI.

DIMENSIONS, ETC., OF PIPES AND PIPE THREADS. (see Fig. 143, next page).

		Sectional area.			Actual weight per foot run lbs.			Contents.		Extl. surface, sq. ft.		Screw threads (Brit. Stan).		
Internal diam.	External diam.	Internal sq. inches.	External sq. inches.	Theoret. area of metal sq. ins. (see weight).	Gas, quality.	Water, quality.	Steam, quality.	Contents of one ft. cub. ins. (theoretical).	Ft. run for one cub. ft.	Of one ft. run sq. ft.	Ft. run for one sq. ft.	Threads per inch.	Depth of thread.	Outside diam. of thread.
¼″	.515	.049	.208	.159	.38	.425	.47	.588	2,940	.134	7.47	19	.0335	.518
⅜″	.685	.111	.369	.258	.58	.64	.7	1.33	1,300	.179	5.59	19	.0335	.656
½″	.844	.196	.559	.363	.81	.9	.97	2.35	735	.229	4.37	14	.0455	.825
¾″	1.063	.442	.889	.447	1.16	1.27	1.4	5.30	326	.277	3.61	14	.0455	1.041
1″	1.34	.785	1.410	.625	1.6	1.78	1.96	9.42	183.5	.344	2.91	11	.058	1.309
1¼″	1.656	1.227	2.150	.923	2.36	2.6	2.84	14.7	117.5	.440	2.28	11	.058	1.650
1½″	1.938	1.767	2.950	1.183	3	3.25	3.5	21.2	82.3	.508	1.97	11	.058	1.882
2″	2.406	3.142	4.535	1.393	4.1	4.5	4.9	37.7	45.9	.622	1.61	11	.058	2.347
2½″	3.063	4.909	7.370	2.461	5.33	5.78	6.32	58.9	29.4	.802	1.25	11	.058	2.960
3″	3.50	7.069	9.621	2.552	6.33	6.85	7.50	84.7	20.4	.907	1.10	11	.058	3.460
4″	4½	12.57	15.90	3.33	8.05	8.72	9.56	150.7	11.5	1.18	.848	11	.058	4.45
5″	5½	19.63	23.76	4.13	10.17	11.01	12.05	235.5	7.35	1.44	.695	11	.058	5.45
6″	6½	28.27	33.18	4.91	12.57	13.60	14.88	339	5.1	1.70	.589	11	.058	6.45

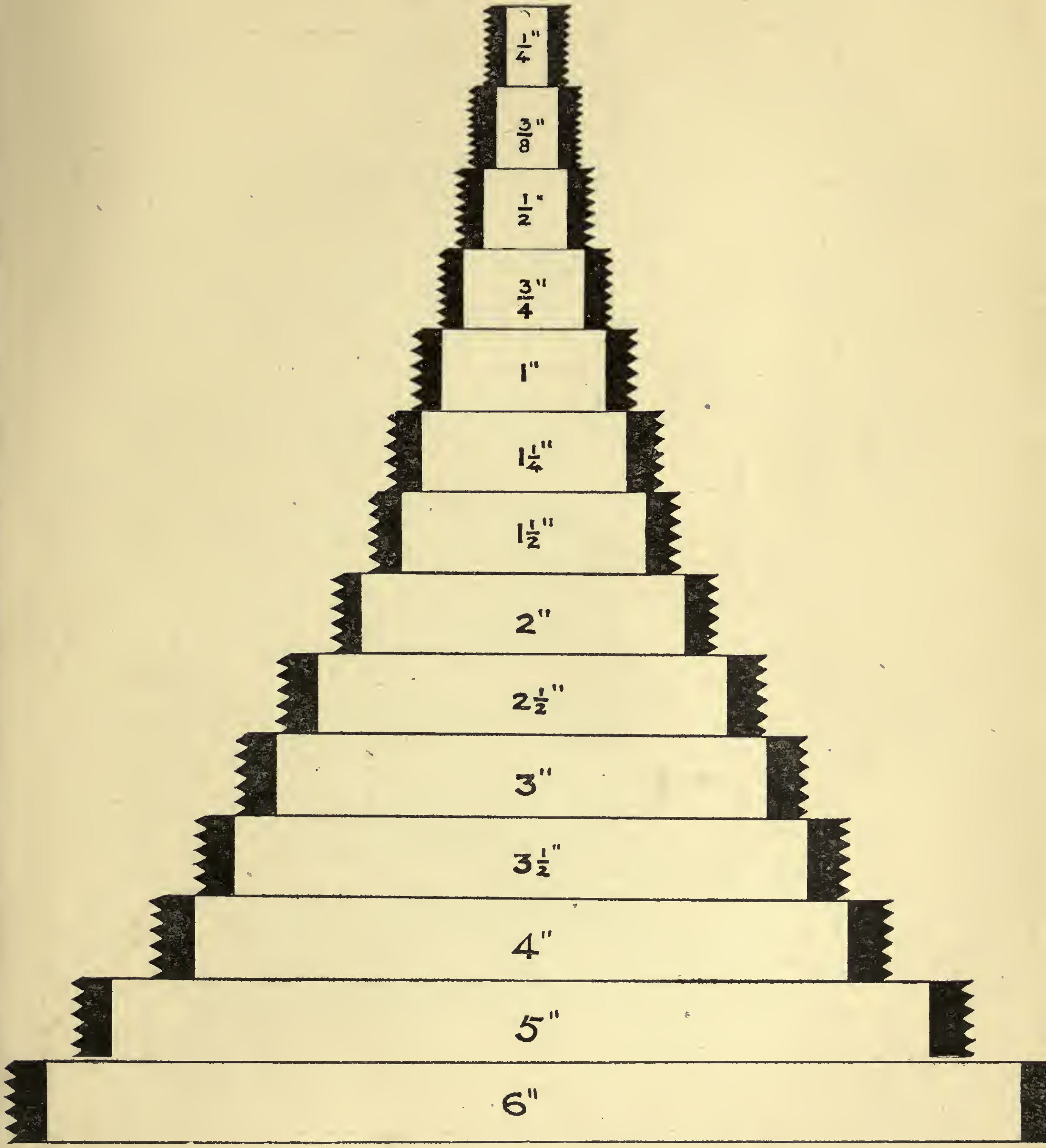


FIG. 143.
Section of Pipe Threads, British Standard.

TABLE XXII. (see Figs. 137, 138).

APPROXIMATE LOSS OF PRESSURE IN UNCOATED LOW PRESSURE STEAM PIPES.

Initial pressure about 40" of water column (coating pipes reduces the loss of pressure from 5% to 10% in ordinary cases).

Dia. of pipe	½"	¾"	1"	1¼"	1½"	2"	2½"	3"	4"	5"	6"
Loss of pressure inches of water per 10ft. run.	B.T.U. delivered at end of pipe per hour.										
0.5	3,000	10,000	20,000	37,000	60,000	125,000	235,000	370,000	420,000	585,000	1,050,000
1	5,000	15,000	30,500	56,000	88,000	182,000	340,000	530,000	620,000	850,000	1,550,000
1.5	6,500	18,500	38,500	70,000	110,000	235,000	410,000	635,000	770,000	1,065,000	1,950,000
2	7,500	22,000	45,000	82,000	128,500	260,000	470,000	735,000	895,000	1,250,000	
2.5	9,000	25,000	50,000	93,000	144,000	290,000	530,000	820,000	1,000,000	1,400,000	
3	10,000	27,500	55,500	102,000	158,000	320,000	580,000	900,000	1,100,000	1,540,000	
3.5	11,000	30,000	60,000	111,000	172,000	350,000	630,000	970,000	1,190,000	1,670,000	
4	11,500	32,000	64,500	119,000	185,000	372,000	672,000	1,040,000	1,275,000	1,790,000	
4.5	12,200	34,000	69,000	127,000	196,500	395,000	715,000	1,100,000	1,360,000	1,910,000	
5	13,000	36,000	72,500	134,000	208,500	415,000	750,000	1,160,000	1,430,000	2,020,000	
5.5	13,500	38,000	76,000	141,000	223,000	435,000	790,000	1,220,000	1,510,000	2,120,000	
6	14,500	40,000	80,000	148,000	232,000	455,000	825,000	1,270,000	1,575,000	2,225,000	
6.5	15,000	41,500	83,500	154,000	244,000	474,000	860,000	1,325,000	1,640,000		
7	15,800	43,000	86,700	160,000	253,000	495,000	890,000	1,375,000	1,710,000		
7.5	16,200	44,500	90,000	165,500	263,000	512,000	925,000	1,430,000	1,770,000		
8	16,800	46,000	93,000	171,500	272,000	530,000	955,000	1,480,000	1,830,000		
8.5	17,500	47,500	96,000	177,000	281,000	550,000	982,000	1,525,000	1,890,000		
9	18,000	48,800	98,500	182,500	290,000	567,000	1,010,000	1,570,000	1,950,000		
9.5	18,500	50,000	101,000	187,500	300,000	582,000	1,040,000	1,615,000	2,010,000		
10	19,000	51,000	104,000	192,500	305,000	600,000	1,065,000	1,655,000	2,065,000		

TABLE XXIII.

COMPOSITION OF COAL GAS IN VARIOUS LOCALITIES.

1 lb. coal yields about 4.5 cubic feet of gas.
1 cubic foot of gas contains as follows by volume :—

Name of Town.	Hydrogen	Marsh Gas CH_4	CO.	Heavy Hydro-carbons.	N.	CO_2.	O_2.
Edinburgh	.3324	.4293	.0661	.1223	.0364	.0035	.0100
Glasgow	.3918	.4026	.0714	.100	.0307	.0029	.0006
Liverpool	.3644	.4428	.0339	.0790	.0610	.0170	.0019
Leeds	.4023	.4274	.0502	.0728	.0432	.0034	.0007
Nottingham ...	.4552	.3966	.0563	.0563	.0251	.0081	.0024
Birmingham ...	.4023	.3900	.0405	.0476	.101	.015	.0036
Bristol	.4457	.4070	.0477	.0458	.0511	none	.0027
London	.4799	.3764	.0375	.0441	.0595	none	.0026
Chemnitz	.5129	.3645	.0445	.0491	.0141	.0108	—
Heidelburg ...	.4400	.384	.0573	.0727	.0423	.0037	—
Producer gas ...	.025	.020	.220	—	.685	.04	—
Dowson gas ...	.1873	.0031	.2507	.0031	.555	—	—

Volume of 1 pound weight about 28 to 35 cubic feet.

General Composition of Rich Coal Gas.

		B.T.U. per pound.
CO 13.8% by weight	=	597
H 23% ,,	=	14,260
C 50.2% ,,	=	7,279
Total		22,136

Total heat of combustion average about 622 B.T.U. per cubic foot.
Minimum for ordinary coal gas about 500 B.T.U. per cubic foot at 60°.
Maximum for rich gas about 750 B.T.U. per cubic foot at 60°.

TABLE XXIV.

PROPERTIES OF FUELS (see Chapter XVIII.).

	WEIGHT.				COMPOSITION %.							
	Specific gravity, solid.	Weight one cubic foot, solid.	Weight one cubic foot, heaped.	Bulk of one ton.	Carbon.	Hydrogen	Nitrogen.	Sulphur.	Oxygen.	Ash.	Moisture.	Total B.T.U. from one pound.
Coal.												
Anthracite, best ...	1.59	99.6	63	36	95	1.5	1	.5	.5	1.5	—	14,700
,, ordinary ...	—	—	—	—	90	3	1.5	1.5	.1	3	—	—
British coal—												
From about	—	—	—	—	84	5	1	2	4	4	—	—
To	—	—	—	—	78	6	1	1	9	5	—	—
Average	—	—	—	—	80.4	5.19	1.21	1.25	7.87	4.05	—	14,320
Coke.												
From	—	—	—	—	97.6	—	—	.85	—	1.55	—	14,150
To	—	—	—	—	84.8	—	—	.8	—	14.4	—	12,300
Average	.74	46	30	75	93.44	—	—	1.22	—	5.34	—	13,550
Petroleum.												
From	—	—	—	—	82	11.2	—	—	.5	—	—	—
To	—	—	—	—	87	14.8	—	—	5.7	—	—	—
Average	.87	—	—	—	84.7	13.1	—	—	2.2	—	—	20,240
Peat.												
Dried	.3 to 1	—	—	—	60	6	4	.3	25	4.7	—	9,951
With moisture		—	—	—	43	43	3	.2	19	3.5	28	7,156
Average	.528	—	—	—	—	—	—	—	—	—	—	—
Wood.												
Dried	—	—	17 to 23	100 to 140	50	6	40	—	1	3	—	7,800
Undried...	—	—			—	—	—	—	—	—	25	—
Gas.												
Lighting	—	—	—	—	58	23	5.8	—	13.2	—	—	22,136

TABLE XXV. (see Chapter XI., p. 230).

Walker's experiments on Propeller Ventilating Fans, Nos. 1 to 10, of 23¾ inches diameter, all revolving *inside* delivery air tube of 24 inches bore, except those marked * revolving 4½ inches out of tube. Blades set at angle to plane of revolution. Anemometer placed at radius of 1⅞, 5⅛, 7⅞, 10⅝ inches from axis of tube.

No. of Fan.	Angle of Blades.	Revolutions per minute.	Air discharged per minute. Cubic feet.	Horse power.		Efficiency.	
				Brake. H.P.	Air. H.P.	Mechanical. P.C.	Volumetric. P.C.
1	17°	890	2,128	.0458	.0097	21.2	38.2
2	17°	860	2,888	.0866	.0241	28	54
3	17°	645	2,172	.0430	.0103	23.9	53
4	17°	525	2,139	.0350	.0098	28	65
5	27°	610	2,790	.0829	.0218	26.3	73
6	27°	490	2,536	.0770	.0164	21.3	82
7	17°	758	2,427	.0650	.0144	22.2	51
8	17°	650	2,330	.0677	.0127	18.8	57
	27°	*600	*3,385	.0894	.0390	43	89.8
	27°	595	2,773	.0742	.0214	29.9	74
	40°	*495	*3,265	.0916	.0350	38.2	105
9	17°	600	1,633	.0590	.0043	7.4	43.3
	27°	600	2,348	.0768	.0130	16.9	62
	27°	*600	*2,965	.0892	.0262	29.4	78
	40°	570	2,870	.1075	.0237	22	80
10	27°	*600	*2,891	.0809	.0243	30	76.7
	17°	*600	*1,675	.0550	.0047	8.5	44.4
11	17°	850	2,720	.0620	.0202	32.6	50.9
	27°	610	2,641	.0458	.0185	40.4	68.9
	27°	*600	*2,699	.0489	.0190	40.4	71.6
12	27°	600	2,222	.0473	.0110	23.3	58.9
13	27°	600	2,737	.0671	.0200	30	72.6
	17°	600	1,926	.0431	.0070	16.7	51
14	17°	604	1,846	.0300	.0063	21	48.6
	40°	605	3,272	.0874	.0348	39.8	86
15	40°	600	3,270	.0862	.0350	40.9	86.7
16	35°	600	2,580	.0510	.0172	33.7	68.4
17	35°	600	2,705	.0538	.0199	37	71.7

TABLE XXV. *(Continued).*

Comparison of Propeller Ventilating Fans, Nos. 16 and 17, with blades set at different angles to plane of revolution. Both fans 23¾ inches diameter, revolving inside delivery air tube of 24 inches bore; one speed, 600 revolutions per minute. Friction of motor included.

Angle of Blades.	FAN NO. 16.					FAN NO. 17.				
	Air discharged per min.	Horse power.		Efficiency.		Air discharged per min.	Horsepower.		Efficiency.	
		Gross.*	Air.	Mechanical.†	Volumetric.		Gross.*	Air.	Mechanical.†	Volumetric.
	Cub. feet.	H.P.	H.P.	P.C.	P.C.	Cub. feet.	H.P.	H.P.	P.C.	P.C.
15°	1,079	.0377	.0012	3.1	28.6	1,562	.0453	.0038	8.3	41.4
20°	1,622	.0429	.0042	9.7	43	1,916	.0519	.0070	13.4	50.8
25°	2,030	.0521	.0084	16.1	53.8	2,291	.0620	.0120	19.3	60.8
27°	2,185	.0588	.0106	18	58	2,423	.0672	.0143	21.2	64.3
30°	2,439	.0698	.0145	20.7	64.7	2,598	.0761	.0176	23.1	68.9
35°	2,580	.0848	.0172	20.2	68.4	2,705	.0876	.0199	22.7	71.7
40°	2,627	.0958	.0182	18.9	69.7	2,805	.1045	.0222	21.2	74.4

* Gross horse-power = brake horse-power + horse-power due to friction of motor.

† Mechanical efficiency = ratio of horse-power in air discharged to gross horse-power.

Experiments on Propeller Ventilating Fan, No. 17, with blades set at different angles to plane of revolution. Fan 23¾ inches diameter, revolving inside delivery air tube of 24 inches bore; speed 600 revolutions per minute. Anemometer placed at radius of 1⅞, 5⅛, 7⅞, 10⅝ inches from axis of tube.

Angle of Blade.	Air discharged per min.	Horse power.		Efficiency.	
		Brake.	Air.	Mechanical.	Volumetric.
	Cub. feet.	H.P.	H.P.	P.C.	P.C.
15°	1,562	.0115	.0038	33	41.4
20°	1,916	.0181	.0070	38.5	50.8
25°	2,291	.0282	.0120	42.2	60.8
27°	2,423	.0334	.0143	42.8	64.3
30°	2,598	.0423	.0176	41.6	68.9
35°	2,705	.0538	.0199	37	71.7
40°	2,805	.0707	.0222	31.4	74.4
45°	2,827	.0858	.0227	26.4	76.7
50°	2,771	.1024	.0214	20.9	75.2
60°	1,534	.0911	.0035	3.8	40.7

TABLE XXVI.

LENGTHS OF SINGLE 1″ PIPE EQUIVALENT IN HEATING EFFECT TO 1 FT. RUN OF PIPES, ETC., OF OTHER SIZES.

Size of pipe ...	⅜″	½″	¾″	1″	1¼″	1½″	2″	2½″	3″	4″	5″	6″
Length equivalent to 1 ft. run of 1″	.53	.72	.83	1	1.23	1.36	1.60	2	2.20	2.76	3.24	3.73

Kind of Surface. 1 square foot of standard 2 column radiator.	Equivalent. Ft. run of 1″ pipe.
1 section	2.15
2 ,, ...	1.86
3 ,, ...	1.80
4-6 ,, ...	1.73
More than 6 sections	1.67

Flat surface 1 sq. ft. = 2.17

TABLE XXVII.

VALUES OF *K* FOR TRANSMISSION OF HEAT THROUGH THIN METAL PLATE FROM AIR TO AIR (AFTER RIETSCHEL).

Diff. temp. deg. F.	18	36	54	72	90	108 and over
Velocity hot air in ft./sec.						
2	.20	.29	.34	.39	.43	.45
4	.35	.48	.56	.60	.63	.65
6	.46	.62	.69	.72	.74	.76
8	.54	.69	.78	.80	.83	.85
10	.62	.76	.84	.88	.90	.92
15	.74	.88	.96	1	1.02	1.05
20	.82	.97	1.04	1.09	1.12	1.14
25	.88	1.03	1.10	1.16	1.18	1.20
30	.92	1.07	1.14	1.20	1.22	1.24
35	.94	1.09	1.16	1.22	1.24	1.26
40	.94	1.11	1.17	1.22	1.24	1.26

TABLE XXVIII.

VALUES OF $24\,\dfrac{D_2 - D_1}{D_2 + D_1}$

These values multiplied by the height (feet) of centre of radiator above centre of boiler give approximate circulating pressure in inches of water (see page 481 and Fig. 120).

Flow Temp.	Return Temp.	$24\frac{D_2 - D_1}{D_2 + D_1}$	Flow Temp.	Return Temp.	$24\frac{D_2 - D_1}{D_2 + D_1}$	Flow Temp.	Return Temp.	$24\frac{D_2 - D_1}{D_2 + D_1}$	Flow Temp.	Return Temp.	$24\frac{D_2 - D_1}{D_2 + D_1}$	Flow Temp.	Return Temp.	$24\frac{D_2 - D}{D_2 + D_1}$
200	195	.025	195	120	.302	185	115	.270	170	150	.081	155	140	.057
	190	.048		115	.317		110	.284		145	.100		135	.074
	185	.072		110	.332	180	175	.022		140	.118		130	.092
	180	.096	190	185	.023		170	.044		135	.136		125	.108
	175	.118		180	.046		165	.065		130	.153		120	.125
	170	.140		175	.069		160	.086		125	.170		115	.140
	165	.161		170	.090		155	.106		120	.187		110	.155
	160	.181		165	.112		150	.126		115	.202	150	145	.018
	155	.201		160	.133		145	.145		110	.217		140	.036
	150	.221		155	.152		140	.163	165	160	.020		135	.054
	145	.240		150	.172		135	.180		155	.040		130	.070
	140	.258		145	.191		130	.198		150	.060		125	.087
	135	.276		140	.210		125	.215		145	.079		120	.104
	130	.293		135	.228		120	.230		140	.097		115	.120
	125	.309		130	.244		115	.246		135	.115		110	.134
	120	.324		125	.261		110	.261		130	.132	145	140	.019
	115	.340		120	.277	175	170	.020		125	.149		135	.037
	110	.354		115	.293		165	.041		120	.165		130	.054
195	190	.024		110	.307		160	.062		115	.180		125	.07
	185	.048	185	180	.022		155	.083		110	.195		120	.087
	180	.070		175	.046		150	.103	160	155	.02		115	.102
	175	.094		170	.068		145	.122		150	.04		110	.116
	170	.115		165	.090		140	.140		145	.059	140	135	.018
	165	.136		160	.110		135	.158		140	.077		130	.035
	160	.157		155	.130		130	.175		135	.094		125	.052
	155	.176		150	.149		125	.192		130	.111		120	.068
	150	.197		145	.168		120	.208		125	.128		115	.084
	145	.216		140	.186		115	.224		120	.145		110	.099
	140	.234		135	.204		110	.238		115	.160			
	135	.251		130	.222	170	165	.020		110	.175			
	130	.269		125	.238		160	.040	155	150	.02			
	125	.285		120	.254		155	.061		145	.039			

TABLE XXIX. (see Fig. 127 and page 510).

BARKER'S METHOD OF DETERMINING APPROXIMATE PIPE SIZES IN HOT WATER INSTALLATION FOR 30° DIFFERENCE FLOW AND RETURN.

INCREASE BY 50% FOR 40° AND BY 110% FOR 50° DIFFERENCE.

Height / Travel	½″	¾″	1″	1¼″	1½″	2″	2½″	3″	4″	5″	6″
.005		830	2000	3700	6150	12500	22500	36000	80000	136000	228000
.006		940	2200	4150	6850	14000	25000	40000	90000	151000	253000
.007		1050	2470	4550	7500	15300	27800	44500	100000	167000	277000
.008		1150	2700	4900	8150	16700	30000	48200	108000	180000	300000
.009		1250	2900	5300	8800	18000	32800	52000	115000	195000	325000
.010		1340	3150	5700	9400	19200	35000	56000	123000	207000	344000
.012		1500	3500	6350	10500	21500	39000	63000	137000	230000	385000
.014		1670	3900	7000	11500	23800	43000	69000	150000	255000	420000
.016		1820	4200	7600	12500	26000	47000	75000	163000	276000	460000
.018		1970	4600	8150	13400	27800	50000	80000	174000	298000	490000
.020		2050	4900	8700	14300	29500	54000	86000	186000	318000	522000
.025		2460	5500	10000	16500	34100	62000	97500	212000	365000	600000
.030	1100	2770	6300	11200	18300	38100	69500	109000	236000	405000	670000
.035	1200	3080	7000	12400	20200	42500	76000	120000	256000	450000	740000
.040	1300	3350	7600	13400	21800	46000	82000	130000	280000	485000	800000
.045	1390	3550	8300	14400	23500	49000	88000	140000	300000	520000	850000
.050	1460	3900	8800	15400	25000	53000	93000	148000	318000	555000	900000
.060	1610	4400	9800	17000	27500	58500	104000	164000	350000	620000	1000000
.070	1760	4850	10800	18800	30000	64000	113000	180000	385000	675000	1100000
.080	1900	5300	11600	20000	33000	69000	123000	194000	415000	730000	1180000
.090	2020	5700	12500	21600	35000	74000	130000	207000	440000	780000	1270000
.10	2160	6150	13400	23000	37500	79000	139000	220000	470000	830000	1340000
.12	2400	6850	15000	25700	41500	88000	154000	245000	520000	920000	1490000
.14	2620	7500	16400	28000	46000	96000	168000	267000	570000	1010000	1630000
.16	2820	8200	17600	30600	49000	105000	182000	290000	615000	1090000	1750000
.18	3050	8800	19000	33000	53000	112000	195000	310000	655000	1160000	1880000
.20	3210	9300	20000	35000	56000	118000	207000	325000	690000	1240000	2000000
.25	3600	10600	22500	40000	64000	135000	234000	375000	790000	1400000	2260000
.30	4080	11900	25000	44000	71000	150000	260000	410000	870000	1550000	2500000
.35	4450	13000	27500	48500	78000	165000	285000	450000	950000	1690000	2750000
.40	4800	14000	30000	52500	84000	178000	307000	485000	1030000	1840000	2950000
.45	5150	15000	32000	56500	90000	192000	330000	520000	1100000	1960000	3150000
.50	5350	16000	34000	60000	95000	202000	347000	545000	1160000	2070000	3350000

TABLE XXX.
HOT WATER HEATING.
RIETSCHEL'S METHOD FOR DETERMINING APPROXIMATE PIPE SIZES (see p. 515).

Section A.—For installations not exceeding 80 ft. long, 40 ft. high, difference of temperature 36° F.

Internal Diameter of Pipe :—	1¼″	1½″	2″	2½″	3″	4″	5″	6″
Height centre boiler to centre of index radiator.								
5	12000	18000	38000	60000	105000	250000	430000	680000
10	18000	28500	58200	100000	170000	400000	680000	1080000
15	23500	36200	76500	130500	215000	520000	890000	1380000
20	28500	43200	91000	160000	250000	620000	1050000	1630000
25	32200	50000	105000	180500	290000	710000	1190000	1840000
30	36000	56000	120000	205000	320000	790000	1330000	2060000
35	38000	62000	125000	225000	345000	860000	1460000	2250000
40	41500	68000	130000	240000	360000	930000	1580000	2420000

Section B.—Installation as Section A, but with 54° F. difference of temperature.

5	23000	37000	78000	140000	210000	520000	900000	1360000
10	37000	58000	122000	220000	330000	810000	1380000	2120000
15	48000	75000	156000	280000	430000	1060000	1760000	2710000
20	58000	90000	187000	340000	510000	1260000	2100000	3240000
25	66000	103000	214000	370000	590000	1440000	2400000	3700000
30	73000	115000	240000	420000	650000	1610000	2670000	4140000
35	80000	127000	260000	460000	710000	1760000	2920000	4500000
40	87000	136000	270000	500000	760000	1900000	3150000	4800000

Section C.—For installations between 80 and 160 ft. long, 40 ft. high, 36° F. difference of temperature.

5	8000	12500	30000	50000	75000	190000	320000	510000
10	12500	20000	44000	75000	120000	295000	500000	780000
15	16500	26500	54500	100000	155000	380000	640000	1010000
20	20000	32500	64500	120000	180000	455000	770000	1220000
25	23000	39000	75000	140000	210000	520000	880000	1400000
30	26000	44500	85000	150000	240000	580000	990000	1560000
35	28000	50500	94000	160000	260000	640000	1090000	1700000
40	32000	56000	105000	170000	280000	690000	1180000	1830000

TABLE XXX. *(Continued).*

Section D.—As Section C, but temperature difference 54° F.

Internal Diameter of Pipe :—	1¼″	1½″	2″	2½″	3″	4″	5″	6″
Height centre boiler to centre of index radiator.								
5	17000	26000	62000	100000	160000	400000	660000	1040000
10	25000	40000	89000	160000	240000	600000	1020000	1580000
15	33000	53000	111000	200000	300000	760000	1280000	2020000
20	40000	65000	132000	240000	360000	910000	1530000	2420000
25	46000	77000	152000	275000	415000	1050000	1750000	2760000
30	53000	88000	171000	300000	460000	1160000	1960000	3100000
35	58000	100000	190000	320000	520000	1320000	2150000	3420000
40	63000	108000	207000	340000	560000	1400000	2320000	3720000

TABLE XXXI. (see p. 515).

HOT WATER HEATING.

BIRLO'S METHOD FOR DETERMINING APPROXIMATE PIPE SIZES FOR RISERS.

Size of pipe :—	⅜″	½″	¾″	1″	1¼″	1½″	2″
Height of centre of radiator above centre of boiler.							
5	900	1700	4600	9000	17000	26000	52000
10	1300	2450	6500	13000	24000	36000	75000
15	1550	3000	7800	16000	30000	44000	92000
20	1800	3500	9100	18000	34000	51000	106000
25	2000	3900	10150	22500	38000	57000	120000
30	2250	4300	11000	23000	41000	63000	131000
35	2400	4650	12000	25000	45000	68000	142000
40	2600	4950	13000	27000	48000	73000	152000
45	2800	5300	13800	28000	51000	77000	160000
50	2900	5550	14500	30000	54000	82000	169000
55	3100	5800	15000	31000	56000	86000	177000
60	3200	6100	16000	32000	59000	89000	185000
65	3350	6300	16700	33000	61000	93000	193000

The area of horizontal mains is determined by adding the areas of the risers.

TABLE XXXII.

NATURAL LOGARITHMS TO BASE e.

$\text{Log}_e\, n = 2.302585 \log_{10} n.$

n	$\log_e n$	n	$\log_e n$	n	$\log_e n$
1	.0000	3.10	1.1314	5.20	1.6487
1.05	.0488	3.15	1.1474	5.25	1.6582
1.10	.0953	3.20	1.1632	5.30	1.6677
1.15	.1398	3.25	1.1787	5.35	1.6771
1.20	.1823	3.30	1.1939	5.40	1.6864
1.25	.2231	3.35	1.2090	5.45	1.6956
1.30	.2624	3.40	1.2238	5.50	1.7047
1.35	.3001	3.45	1.2384	5.55	1.7138
1.40	.3365	3.50	1.2528	5.60	1.7228
1.45	.3716	3.55	1.2669	5.65	1.7317
1.50	.4055	3.60	1.2809	5.70	1.7405
1.55	.4383	3.65	1.2947	5.75	1.7492
1.60	.4700	3.70	1.3083	5.80	1.7579
1.65	.5008	3.75	1.3218	5.85	1.7664
1.70	.5306	3.80	1.3350	5.90	1.7750
1.75	.5596	3.85	1.3481	5.95	1.7834
1.80	.5878	3.90	1.3610	6	1.7918
1.85	.6152	3.95	1.3737	6.10	1.8083
1.90	.6419	4	1.3863	6.20	1.8245
1.95	.6678	4.05	1.3987	6.30	1.8405
2	.6931	4.10	1.4110	6.40	1.8563
2.05	.7178	4.15	1.4231	6.50	1.8718
2.10	.7419	4.20	1.4351	6.60	1.8871
2.15	.7655	4.25	1.4469	6.70	1.9021
2.20	.7885	4.30	1.4586	6.80	1.9169
2.25	.8109	4.35	1.4701	6.90	1.9315
2.30	.8329	4.40	1.4816	7	1.9459
2.35	.8544	4.45	1.4929	7.20	1.9741
2.40	.8755	4.50	1.5041	7.40	2.0015
2.45	.8961	4.55	1.5151	7.60	2.0281
2.50	.9163	4.60	1.5261	7.80	2.0541
2.55	.9361	4.65	1.5369	8	2.0794
2.60	.9555	4.70	1.5476	8.20	2.1041
2.65	.9746	4.75	1.5581	8.40	2.1282
2.70	.9933	4.80	1.5686	8.60	2.1518
2.75	1.0116	4.85	1.5790	8.80	2.1748
2.80	1.0296	4.90	1.5892	9	2.1972
2.85	1.0473	4.95	1.5994	9.25	2.2246
2.90	1.0647	5	1.6094	9.50	2.2513
2.95	1.0818	5.05	1.6194	9.75	2.2773
3	1.0986	5.10	1.6292	10	2.3026
3.05	1.1151	5.15	1.6390		

TABLE XXXIII.

RESISTANCE OF AIR FILTERS.

The resistance of air filters (in feet of air column of the same density as the air passing through the filter) is proportional to the quantity of air passing per square foot per second (or in other words, to the first power of the velocity) and to a constant factor p, depending on the permeability of the material employed.

Resistance in feet of air column $= p\ v$.

$$\text{or in inches of water column} = p\ v \times \frac{D_a \times 12}{D_w}$$

$$= p\ v \times \frac{12 \times 0.0863}{62.4\ (1 + a\ t)}$$

$$= .0166 \times \frac{p\ v}{(1 + a\ t)}$$

VALUES OF p.

Nessel cloth (60 threads per inch) $p = 0.0015$ to $.002$.

Moller filter cloth (rough hairy cloth, 40 threads per inch) $p = .024$ to $.03$.

The value of p in other cases may be determined by special experiment. Owing to the immense variety of materials used for this purpose it cannot be given generally.

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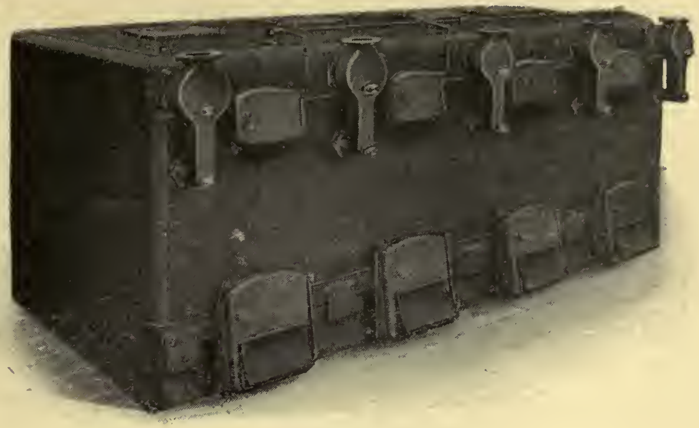

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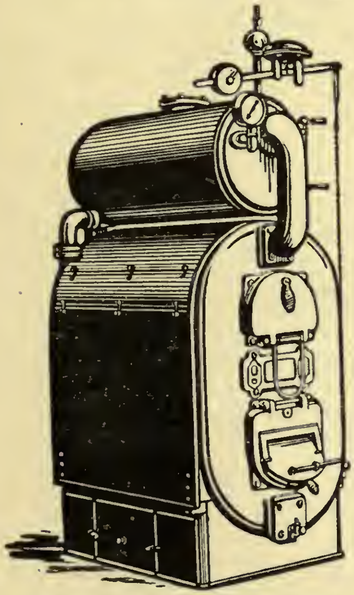

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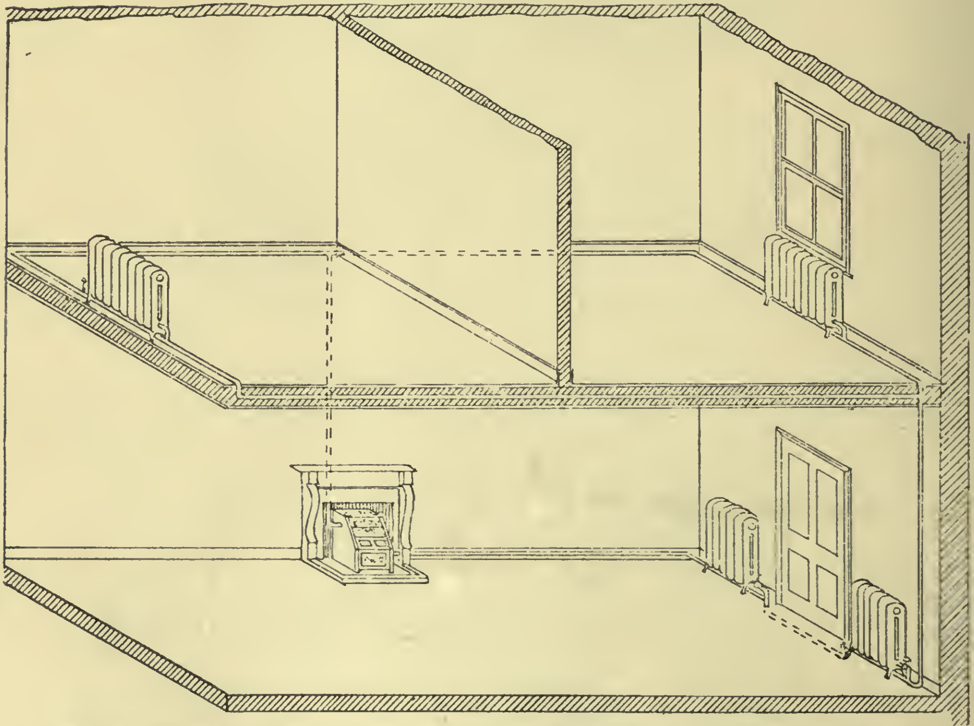

This illustration shows the Hydro-Radiant System of warming a small house. The heat usually wasted up the chimney from the ordinary open fire is used to heat the radiators.

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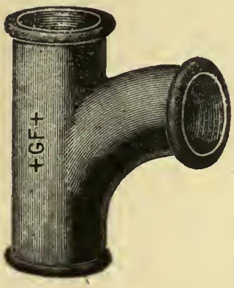

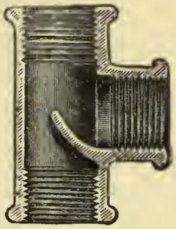

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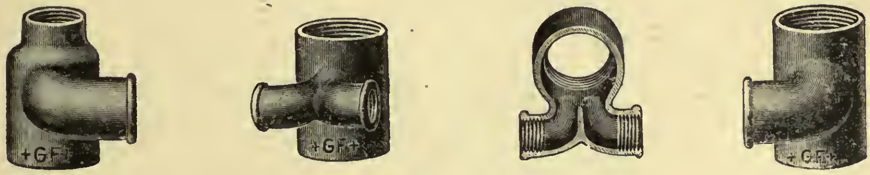

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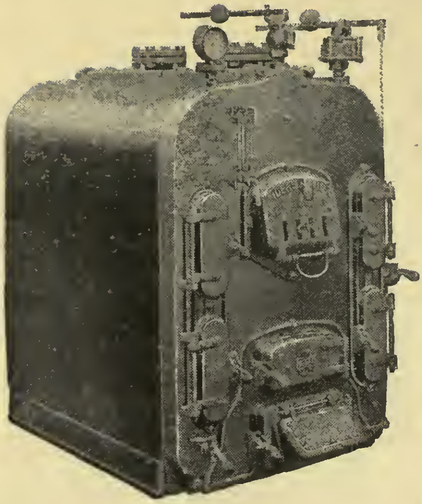

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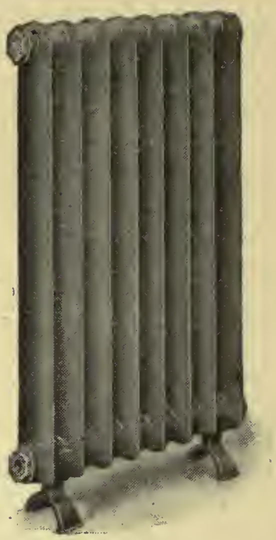

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